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CONTENTS

Page 3: MODERN TRENDS IN ARCH DAM CONSTRUCTION AND DESIGN, *by Dr. Charles Jaeger, Dr. es Sc. tech., Consulting Engineer, Hydraulic Department, The English Electric Company, and Special Lecturer, Imperial College, London.*

29: THE ELECTRIFICATION OF WORSTED MILLS IN THE MIDLANDS, *by J. C. Edmed, D.F.H., A.M.I.E.E., Textile Division, and J. E. Cole, Manager, Leicester Office.*

38: DEPTH INDICATORS FOR MINE HOISTS, *by H. Dolan, A.M.I.Mech.E., A.M.I.E.E., Chief Engineer, Mining Division, and A. Shaw, B.Sc.(Eng.), Mining Division.*

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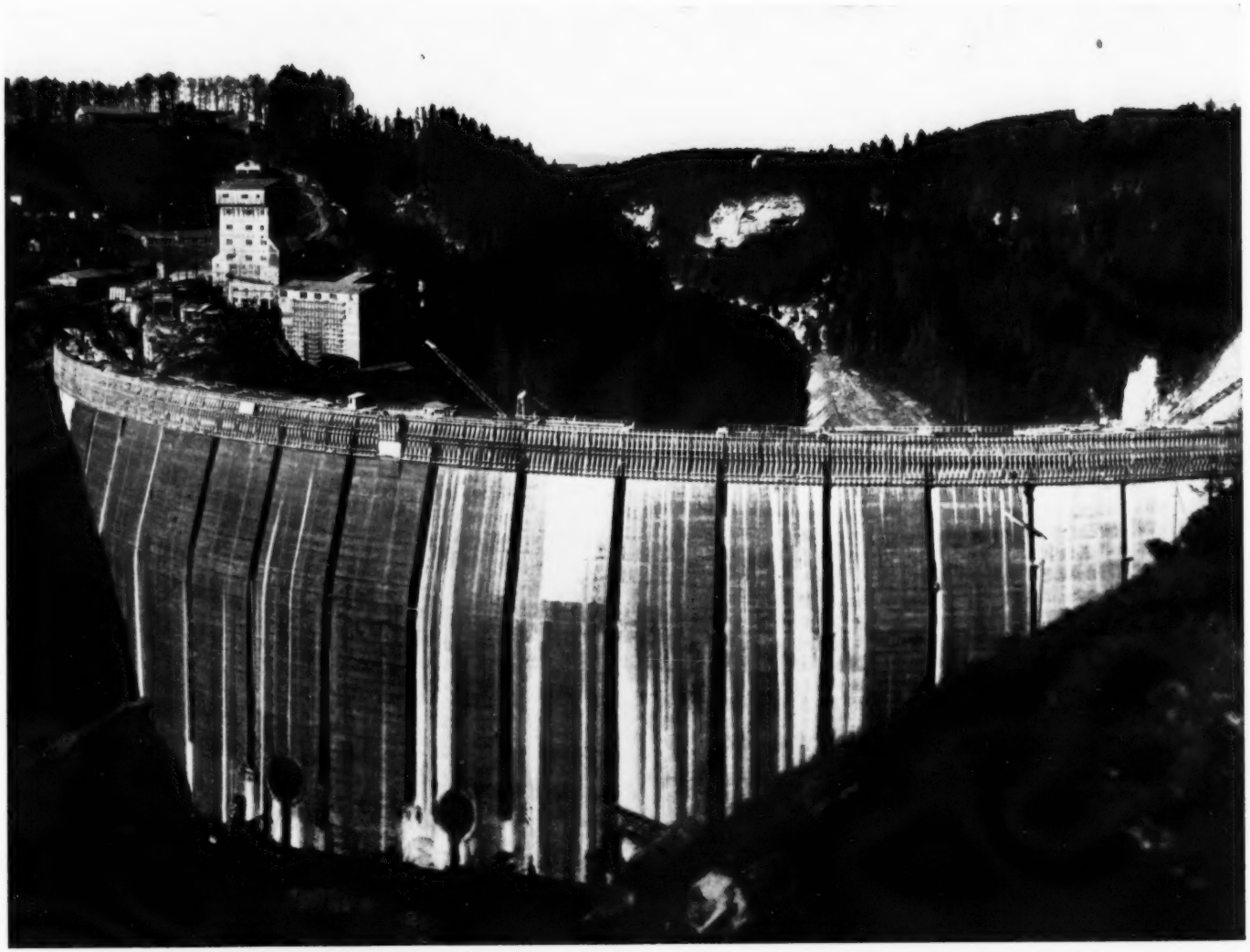


The Rossens dam, Switzerland, as seen from the upstream side during construction.

Modern Trends in Arch Dam Construction and Design

By DR. CHARLES JAEGER, Dr. es Sc. tech., Consulting Engineer, Hydraulic Department, The English Electric Company, and Special Lecturer, Imperial College, London.

This article was originally published in the Civil Engineering and Public Works Review from April to September 1950, *and is reprinted in abbreviated form herein by kind permission of that journal.*

THE THIRD CONGRESS on Large Dams, which was held in Stockholm, Sweden, in 1948, clearly showed divergent trends in arch dam calculation and design. The high degree of responsibility involved in any dam design makes it highly desirable for these differences in opinion to be clearly outlined and the facts on which they are built to be analysed.

One group of engineers emphasises the remarkable achievements of the so-called "trial load method" used in the United States. Another group, however, denies the value of this method which, according to them, is not corroborated by facts and other, alternative, methods of arch dam analysis are proposed.

It is the author's intention to summarise the different points of view and facts as accurately as possible and try to comment upon them.

Fig. 1. *Aerial view of the Rossens dam*

The Rossens dam, Switzerland, as seen from the upstream side during construction.

Trial Load and Similar Methods

The basis for the trial load and similar methods was laid down in the work of H. Ritter. According to them, the dam is assumed to be composed of a series of imaginary horizontal arch slices and another series of imaginary vertical cantilevers (Fig. 2). The elastic grid of arch and cantilever elements thus obtained may be treated as a statically highly indeterminate main system by the known rules of statics. The arches and cantilevers are first considered separately and unknown forces X_i are applied to each point of intersection "i." The unknown forces X_i are determined by means of the equations of elasticity. Finally the deformations and distribution of stresses in the dam can be computed.

H. Ritter made his calculations with one cantilever at the centre of the dam only, but the number of cantilever elements can, of course, be increased by additional sections. The

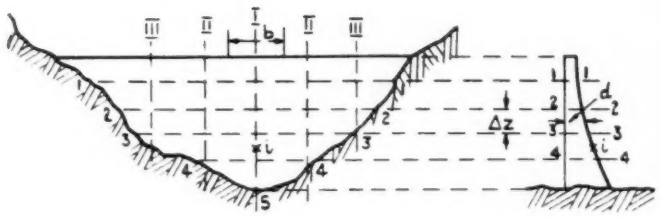


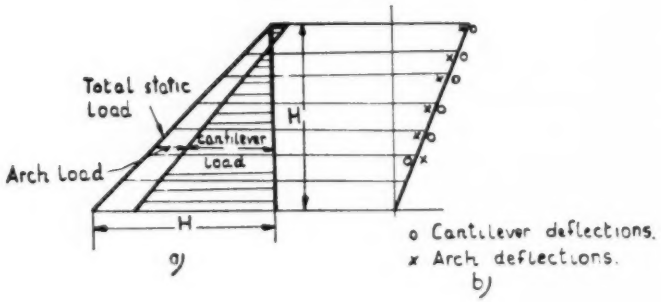
Fig. 2

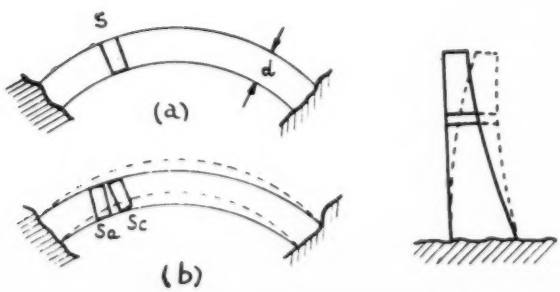
whole computation of the arch and cantilever elements is based on the ordinary bending theory of beams and neglects their resistance to twist.

In following Ritter's idea, Fred A. Noetzli in 1921 introduced in the United States and A. Stucky introduced in 1922 in Europe, a trial load method which, in principle, makes use of the same statically indeterminate main system, but replaces the solution of the equations of elasticity and determination of the unknown forces X_i by a tentative division of the water load between the arch and cantilever elements in such a way that the deflections of both these elements at their points of intersection become equal. A plausible load division for the arches and cantilevers is assumed (Fig. 3a) and the lines of radial deflections are determined separately for either system (Fig. 3b). Naturally, the first trial does not produce equal deflections at the conjugate points. For this reason the division of load is then varied until a sufficiently exact agreement of these deflections is reached. The trial load method is, as it were, a solution of the equations of elasticity by means of a method of iteration and does not differ fundamentally from Ritter's method.

The trial load method, however, is amenable to a progressive amplification which produces a better agreement between the calculated distribution of stresses and deformations and those prevailing in the real shell. The effects of bending moments, or thrust, and of horizontal shear are allowed for when determining the radial deformation of the arch. Further, due consideration is given to the concerted action of the arch and cantilever elements in the dam as a whole. Not only the radial deflections, but the tangential deflections as well as the twisting of arch and cantilever elements at their points of intersection, are made to agree.

Above: *Fig. 3*
Below: *Fig. 4*



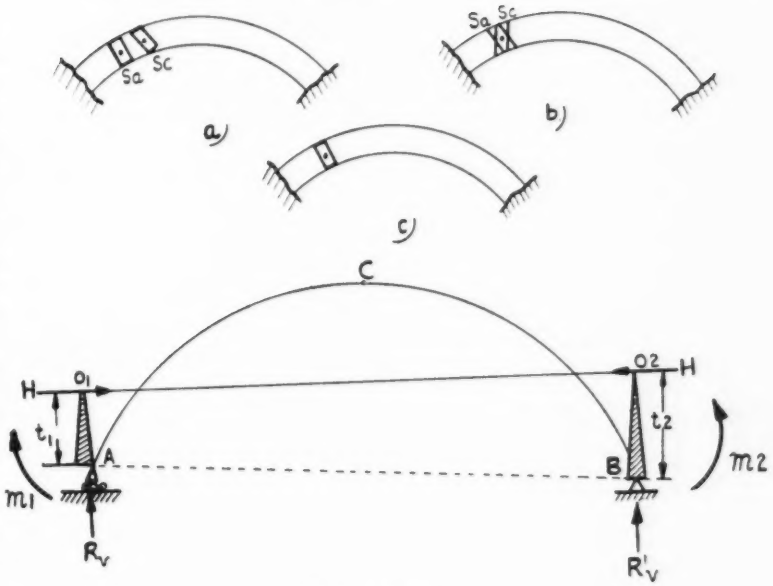


In Fig. 4a is shown a plan of a typical arch which is intersected by a cantilever element at "S"; this is for the unloaded condition. In Fig. 4b external loads have been placed on the dam and the first trial load distribution made between arch and cantilever elements. The cantilever has moved radially to S_c while the arch movement has a tangential and a rotational component, bringing the arch section to S_a.

In Fig. 5a the radial adjustment has been completed. Although radial deflections are equal, it is evident that the arch and cantilever are out of alignment, both tangentially and

in rotation. The first is corrected by the tangential adjustment (Fig. 5b), in which a tangential load is applied to the cantilever, displacing it to the left, and an equal and opposite load applied to the arch. The loads are proportioned by trial to bring the centres of the two common sections together as in Fig. 5b.

Next a twisting moment is applied to the cantilever and an opposing couple of equal magnitude applied as a bending moment on the arch. The magnitude of the couple is adjusted until the two common sections are in rotational agreement, as in Fig. 5c. This is the twist adjustment. The trial load adjustment is now complete and Fig. 5c represents the loaded condition of the dam with the common sections again congruent as they were in Fig. 4a. In practice, the latter adjustments throw the structure out of radial adjustment, so that a radial readjustment is necessary.



Above: Fig. 5
Below: Fig. 6

The use of the trial load method enables the designing engineer to analyse load distributions, deflections and stresses in curved concrete dams of all sizes and shapes, whether of the massive arched gravity type or the relatively thin monolithic arch type.

This method has been dealt with in many papers, most extensively by the United States Bureau of Reclamation and in several handbooks on applied hydraulics and dam design, so that a more detailed analysis of the method can be dispensed with.

Influence of Yielding of Abutments

In all calculation methods, the question of the elastic yielding of the rock in the foundation plays an important part. The theoretical basis for this problem was established by the fundamental investigations of Boussinesq (1885) concerning the elastic half space. Boussinesq calculated the vertical and horizontal displacements of any point of the elastic half space yielding under the pressure of an isolated vertical load. Cerrutti in 1888 and F. Vogt in Trondheim in 1925 investigated the problem of shear forces acting in parallel with the surface of the half space. The rock against which the dam rests may be considered to extend infinitely in comparison with the dam, thus permitting the half-space theory to be applied. According to this theory, the elastic yielding of the rock was computed with regard to bending moments, normal thrust, twist and shear (for the base planes of the arches as well as of the cantilevers), taking into account the inclination of the rock faces toward the two planes. Tonini published a paper covering all these conditions.

With H. Ritter's method and the trial load method this yielding of the foundation may be taken into account by determining the coefficients for the elastic displacement of the unit element due to a normal thrust, a shear, a bending moment and a twist. The determination of deflections of the arch and cantilever elements in the case of an elastic foundation is a simple matter of statics, when the coefficients mentioned are known. One of the pre-requisites for the correct determination of the yielding of the foundation is the knowledge of the modulus of elasticity of the bedrock, and

this may be considerably lower than the modulus of elasticity of the rocks as determined in the laboratory. This point will be discussed later on.

The Simplified Method of M. Ritter

By means of the trial load method or, alternatively, by means of an extended method of H. Ritter, a fairly exact theoretical determination of the distribution of stresses may be expected, though on the assumption of a material idealised in the usual sense (ignoring the effects of shrinkage, flow, possible lack of homogeneity) and of homogeneous foundations. Its use, however, presupposes considerable experience and involves an uncommonly large amount of calculation with extensive tabulation of deformations. Even reducing the scope of the calculations to the initial proposal of H. Ritter and using a grid with only one cantilever in the dam axis and assuming rigid abutments, the calculation of an arch dam is a major task, requiring many months of work.

The basic idea of the method of Prof. M. Ritter consists in finding a convenient static fundamental system so that the equations for the dam calculation can be linearised. At a first stage of the calculation M. Ritter considers only one cantilever and calculates the radial deflections in the direction of the dam axis, but it is possible to introduce several cantilevers at a later stage of the calculation. Elastic yielding of the arch and cantilever foundations are in each case introduced as boundary conditions. Supposing the dam has been reduced to n " arch slices," then the final calculation consists in solving n linear equations with n unknowns $X_1 \ldots X_2 \ldots X_n$. There is no need for any trial and error calculation.

The advantage of M. Ritter's method is that it is far shorter than the usual method of trial and error and includes the elastic yielding of the foundations, which is usually dropped at a first stage of the trial load calculations. The ease with which the elastic boundary conditions are introduced is a main point in favour of M. Ritter's method.

As this method has never been published before and as such a publication is unlikely to be made because of the death of Prof. Ritter, an endeavour has been made to reconstruct it, as follows.

A. General Remarks on Arch Calculation.

Let us consider (Fig. 6) the built-in arch ABC. In order to calculate such an arch, M. Ritter introduces a fundamental system consisting of a simply supported beam. The end A of the beam is rolling free. R_v and R'_v are the vertical reactions in this fundamental system. The unknown quantities to be introduced in the calculation are the arch thrust H and the moments M_1 and and M_2 at the abutments A and B. The points O_1 and O_2 on which the thrusts H are acting are supposed to be rigidly connected to the end cross sections A and B of the arched beam. If the abutments are supposed to be non-elastic, the rotations α and β of the abutments A and B are equal to zero ($\alpha=0$ and $\beta=0$). Likewise, the distance AB remains unchanged and the relative displacement δ of A and B is also equal to nought ($\delta=0$).

In the more general cases where the abutments are elastic, the equations of elasticity may be written thus:

$$\alpha = \alpha_0 + M_1 \alpha_1 + M_2 \alpha_2 + H \alpha_3$$
$$\beta = \beta_0 + M_1 \beta_1 + M_2 \beta_2 + H \beta_3$$
$$\delta = \delta_0 + M_1 \delta_1 + M_2 \delta_2 + H \delta_3$$

In these equations, α_0 is the rotation of the end A of the arch, due to external load only. α_1 is the rotation due to a moment $M_1=1$ acting alone, α_2 to a moment $M_2=1$ acting alone and α_3 to an arch thrust $H=1$ acting alone. The same applies for $\beta_0, \beta_1, \ldots \delta_2, \delta_3$. If the abutments are elastic, then α and β are not equal to nought and have to be calculated as resulting from the yielding of the rock.

The distances $t_1=O_1A$ and $t_2=O_2B$ should be chosen in order to simplify the group of the three equations of elasticity. The values of t_1 and t_2 may be different when calculating steady loads or variable loads.

In the case of steady loads M. Ritter introduces the procedure shown in Fig. 7. The fundamental system introduced is still the simply supported beam.

Let us consider the pressure line of a three pinned arch ACB as shown in Fig. 7. The ordinate of a point P on this pressure line is $v + y^1$, where $y^1 = y + t$ is the ordinate of the arch centre line. The thrust at the abutment A of a three pinned arch is H_0, which is different from the hyperstatic

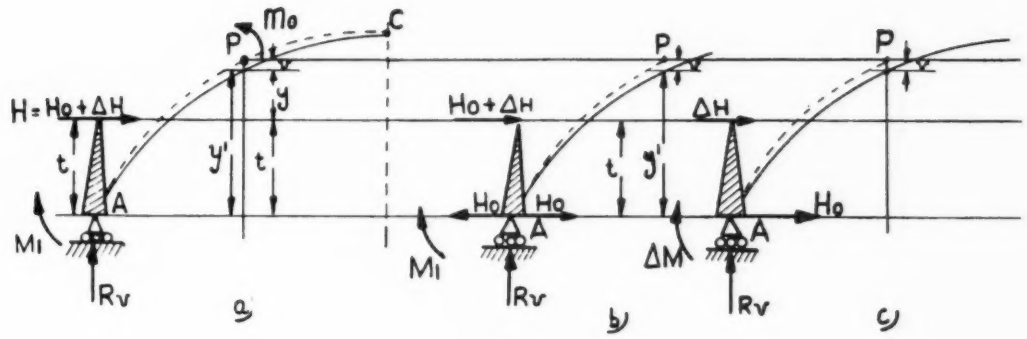


Fig. 7

thrust H at point O of the real arch. Thus (Fig. 7a):—

$$H = H_0 + \Delta H.$$

Let M_0 be the moment of the external loads and reactions acting on the singly supported beam ACB at the right of point P. We can then write, for the three pinned arch ACB:

$$M_0 - H_0(y^1 + v) = o.$$

We now introduce (Fig. 7b) two equal and opposite forces H_0 at A and reduce the forces and moments acting on the real arch to (Fig. 7c):

the thrust $\Delta H = H - H_0$ acting on point O
the thrust Ho acting on the abutment A
the vertical reaction R_v in the simply supported beam ACB, and
the moment $\Delta M = M_1 + H_0 t$.

The moment $H_0 (y^1 + v)$ is always compensated by the moment M_0 of the external forces acting at the right of any point P including the moment of the vertical reaction R'_v.

B. Calculation for an Arch under Uniform Normal Pressure.

Let us consider a symmetrical arch under uniform normal load p (Fig. 8). With such a load, the pressure line of the three pinned arch ACB is a circle. We assume that this pressure line is also the arch centre line so that $v = o$. At any point P of the circle, the forces and moments in the real arch are: (with $M_0 - y^1 H_0 = O$):—

$$\begin{aligned} M &= M_0 - y^1 H_0 + \Delta M - \Delta H y \\ &= \Delta M - \Delta H y \\ N &= N_0 + \Delta H \cos\varphi \\ Q &= -\Delta H \sin\varphi \end{aligned}$$

At point P the sum total M_0 of the moments produced by all external forces (load p) is balanced by the moment $H_0(y+t) = H_0 y^1$. There is no shear force on P, only a constant axial thrust N_0 produced by the load p in the three pinned arch.
At the abutment A, we shall have:—

$$\left.\begin{aligned} M_a &= \Delta M + \Delta H t \\ N_a &= N_0 + \Delta H \cos\Psi \\ Q_a &= -\Delta H \sin\Psi \end{aligned}\right\} \quad \ldots\ldots \quad (1)$$

φ and Ψ being the angles as indicated in Figs. 8a and 9.

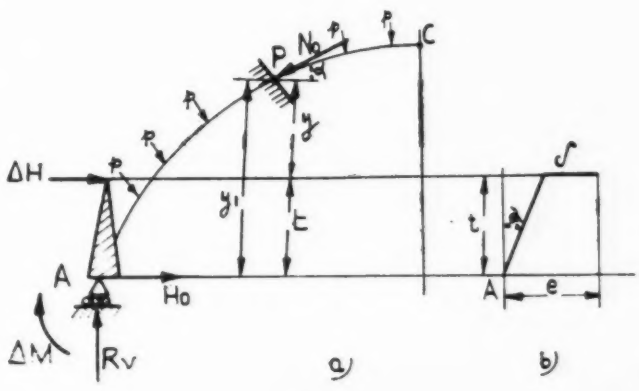


Fig. 8

According to the classical formula of Bresse:

$$\alpha = \int_0^{\Psi} \frac{M\,ds}{E\,I}$$

$$\delta = \int_0^{\Psi} \frac{M\,y\,ds}{E\,I} - \int_0^{\Psi} \frac{N\cos\varphi\,ds}{E\,A} + \int_0^{\Psi} \frac{Q\sin\varphi\,ds}{G\,A^1} - \int_0^{\Psi} \varepsilon_s \cos\varphi\,ds \quad (2)$$

α being the angular rotation of the abutment and δ the horizontal component of the lengthening of the arch due to the elasticity of the arch, the crown C of the arch being supposed fixed. From Fig. 8b is given geometrically:

$$e = \alpha t + \delta \quad \ldots\ldots (3)$$

where e is the horizontal displacement of point A. E is the modulus of elasticity of the concrete and $G = \frac{E}{2\,(1 + 1/m)}$, with m = coefficient of Poisson, I, A and $A^1 = \frac{m}{1+m}\,A \cong \frac{5}{6}\,A$ referring to the arch cross section (if d and Δz are the sides of the cross section, $A = d\,\Delta z$, $I = \frac{d^3\,\Delta z}{12}$ with $\Delta z = 1$). ε_s is a coefficient representing the shrinkage of the concrete or strain due to temperature variations.

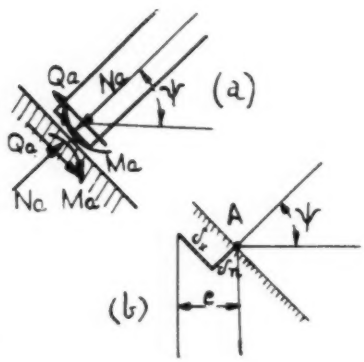


Fig. 9

The deformations α and δ of the arch are equal to the deformations α and δ of the abutment due to M_a, N_a and Q_a (Fig. 9a) so we can write:

$$\alpha = -\varepsilon Ma + \varepsilon' Qa$$
$$= -\varepsilon\,\Delta M - (\varepsilon t + \varepsilon' \sin\Psi)\,\Delta H$$

ε and ε' being coefficients of elastic rock deformation which may be equated to (Vogt or Ritter):

$$\varepsilon \cong \frac{0.6d}{E_f\,I} \text{ and } \varepsilon' \cong \frac{0.6}{E_f\,I}$$

Where E_f is the modulus of elasticity of the foundation rock. δ_n is the rock deformation in direction of the thrust Na and δ_x is the yielding of the rock perpendicular to Na.

Introducing the following simplified notations,

$\delta^1_N = \delta_N \cos\Psi$, $\delta^1_M = \delta_M \sin\Psi$ and $\delta^1 = (\delta_N \cos^2\Psi + \delta_M t \sin\Psi + \delta_Q \sin^2\Psi)$

and equating the elastic deformations of the arch and the deformations of the abutments, we find:

$$\left.\begin{aligned}
&\Delta M\left(\int_0^{\Psi}\frac{ds}{E\,I} + \varepsilon\right) - \Delta H\left(\int_0^{\Psi}\frac{y\,ds}{E\,I} - \varepsilon t - \varepsilon^1 \sin\Psi\right) = 0\\
&\Delta M\left(\int_0^{\Psi}\frac{y\,ds}{E\,I} - \delta^1_M - \varepsilon t\right) - \Delta H\left(\int_0^{\Psi}\frac{y^2 ds}{E\,I} + \int_0^{\Psi}\frac{\cos^2\varphi\,ds}{E\,A} + \int_0^{\Psi}\frac{\sin^2\varphi\,ds}{G\,A^1}\right)\\
&+\delta^1 + \varepsilon t^2 + (\varepsilon^1 t \sin\Psi)\delta^1_N = N_0\left(\int_0^{\Psi}\frac{\cos\varphi\,ds}{E\,A} + \delta^1_N\right)\\
&\qquad + \int_0^{\Psi}\varepsilon_s \cos\varphi\,ds
\end{aligned}\right\} \quad (6)$$

In these equations we are still free to choose the distance t and this can be done so that

ΔM becomes equal to nought‡:

$$\Delta M = o, \quad \ldots\ldots (7)$$

if we write that in the first equation (6):

$$t = \frac{\int_0^{\Psi}\frac{y^1 ds}{E\,I} - \varepsilon^1 \sin\Psi}{\int_0^{\Psi}\frac{ds}{E\,I} + \varepsilon} \quad \ldots\ldots (9)$$

Then: $\Delta H =$

$$\frac{-N_0\left(\int_0^{\Psi}\frac{\cos\varphi\,ds}{E\,A} + \delta^1_N\right) + \int_0^{\Psi}\varepsilon_s\cos\varphi\,ds}{\int_0^{\Psi}\frac{y^2 ds}{E\,I} + \int_0^{\Psi}\frac{\cos^2\varphi\,ds}{E\,A} + \int_0^{\Psi}\frac{\sin^2\varphi\,ds}{G\,A^1} + \delta^1 + \varepsilon t^2 + \varepsilon^1 t \sin\Psi} \quad (10)$$

or, more simply:

$$\Delta H = -C_1 No + C_2 \quad \ldots\ldots (11)$$

If we take $\varepsilon_s = o$ then

$$\Delta H = -C_1 No \quad \ldots\ldots (11a)$$

C_1 and C_2 are constant values.

‡ The direction of Q_a in Fig. 9 should be reversed when considering equation (1).

Thanks to these very simplified values $\Delta M = o$ and ΔH = linear function of N_0 the calculation of the elastic displacement δ_s of the crown of the arch is a straightforward one. δ_s will depend on $M = \Delta M - \Delta H y$, $N = No + \Delta H \cos \varphi$ and $Q = -\Delta H \sin \varphi$ and, therefore, is finally a linear function of N_0 alone. N_0 being a linear function of the load, the elastic displacement δ_s is finally a linear function of the load on the arch.

If the arch is a circular one, then all the formulae become simplified in the usual manner.

C. Calculation of Cantilever Deflections.

The calculation of the cantilever deflections is a straightforward one. Let us suppose (Fig. 11) that a force $P_i = 1$ is working at point i. We want to know the deflection produced at point j by this force $P_i = 1$ in i. Let us suppose that element ds alone is elastic. The angular deflection of this element is:

$$d\varphi = \frac{M\, ds}{E\, I}$$

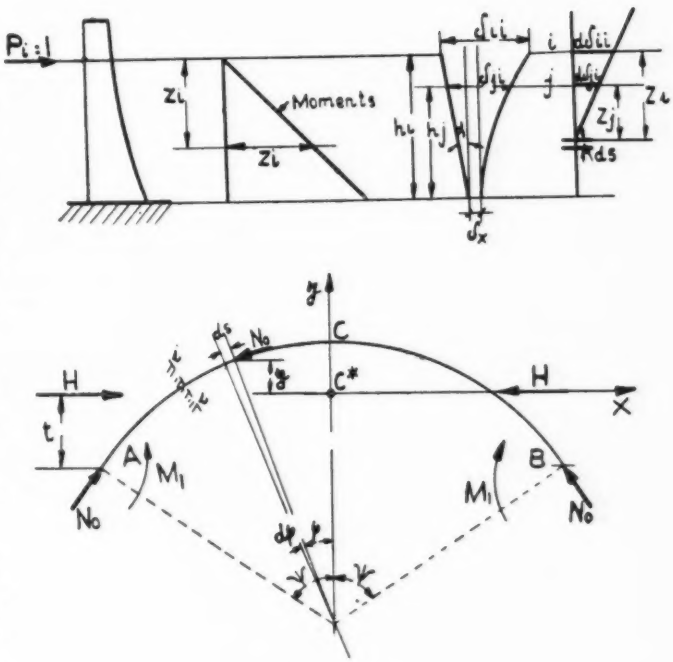
M = Moment at point ds due to $P_i = 1$; $M = z_i$,
E = Modulus of elasticity,
I = Moment of inertia of the cantilever considered at point ds. Owing to $d\varphi$ the elastic displacement of point j is:

$$d\, \delta_{ji} = z_j\, d\varphi = \frac{z_i\, z_j\, ds}{E\, I}$$

The total displacement of point j, owing to $P_i = 1$ in i is obtained when considering that all elements ds between the foundation level and the level of j are elastic. We, therefore, obtain for this displacement:

$$\int \frac{z_i\, z_j\, ds}{E\, I}$$

To this elastic displacement we have to add the tangential displacement δx of the foundations and the displacement αh_j due to the angular rotation α of the foundations; so that:



Above: *Fig.* 11
Below: *Fig.* 12

$$\delta_{ji} = \int \frac{z_i\, z_j\, ds}{E\, I} + \delta_x + \alpha\, h_j$$

All the values δji have to be calculated for the cantilever.

D. Total Elastic Deformation of the Grade System.

Let us consider the deflections of the dam crown only. The dam is assumed to be composed of n "arch slices" in the usual manner, the height of any slice being Δz_i. The width of the cantilever at the crown is b. The arches are loaded with uniform water pressure, the total water loads are:

$W_1, W_2, \ldots . W_i \ldots . W_n$

Part of these loads are taken by the cantilevers, which will be under the action of the forces:

$P_1, P_2, \ldots . P_i \ldots . P_n$

so that the loads on the n arches are:

$(W_1 - P_1), (W_2 - P_2) \ldots . (W_i - P_i) \ldots .$
$(W_n - P_n)$

Let us write that:

δ_{ji}
is the elastic deflection of point j of the central cantilever produced by a force:

$P_i = 1$

acting at point i.
Furthermore, let:

δ_{si}

be the deflection of the crown of arch i due to the uniform load $(W_i - P_i)$ $= 1$ acting on arch i. We have seen that the crown deflection is a linear function of the load, so we can equate the final deflection to

$$(W_i - P_i)\ \delta_{si}$$

The equations of elasticity are obviously:

$$P_1\,\delta_{11} + P_2\,\delta_{12} + P_3\,\delta_{13} + \ldots\ldots = (W_1 - P_1)\,\delta_{s1}$$
$$P_1\,\delta_{21} + P_2\,\delta_{22} + P_3\,\delta_{23} + \ldots\ldots = (W_2 - P_2)\,\delta_{s2}$$
$$P_1\,\delta_{31} + P_2\,\delta_{32} + P_3\,\delta_{33} + \ldots\ldots = (W_3 - P_3)\,\delta_{s3}$$

We know all the factors δ_{s1}, δ_{s2}, δ_{s3} . . . and δ_{11}, δ_{12}, δ_{13} . . . δ_{ji} . . . the n forces P_1, P_2 . . . P_n are obtained from the solution of the n linear equations of elastic deformation.

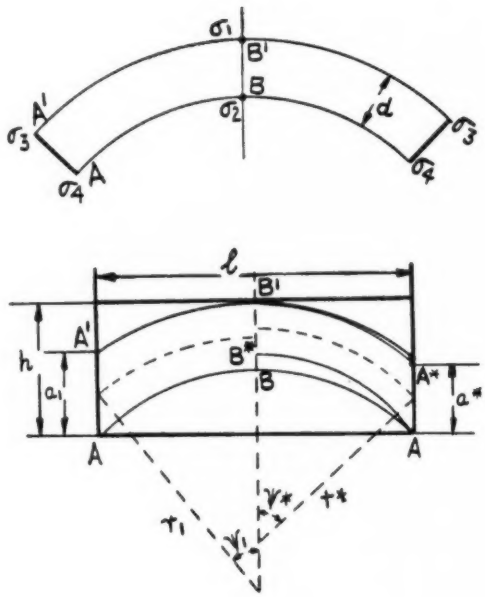
The main point in this method is that, thanks to a convenient choice of the fundamental system giving $M_0 - H_0 y = O$ and $\Delta M = O$, the deflection factors δ_{si} are shown to be functions of the sole thrust N_{oi} produced by the arch load $(W_i - P_i) = 1$. The method is a straightforward one, especially for circular arches.

It needs to be emphasised that the method of Ritter includes the yielding of the abutments, which can be considered even at an early stage of the dam analysis. Recognition of this point is highly important for further comparison of different methods.

This method which was contemplated for use mainly as a first analysis for arch dams could be elaborated for more advanced arch dam analysis on the lines of the American trial load method.

Other Proposed Calculation Methods

A considerable number of experts are opposed to the use of the "trial load" and other similar methods for the calculation of arch dams. They claim that the theory leads to calculations which are too complicated or, alternatively, which are inaccurate owing to neglect of the tangential adjustment and the effect of torsion.



Above: Fig. 13
Below: Fig. 14

A. THE METHOD OF EFFECTIVE ARCHES (Figs. 12, 13 and 14)

One of the alternative methods proposed by French engineers is that of "effective arches" (voûtes actives). Let us suppose that the arch is circular and symmetrical, with constant thickness d, r being the radius of the centre line. Let C* be the centre of elasticity of the arch, which is given by:

$$y_1 = \frac{\int_{-\Psi}^{+\Psi} y^*\,ds/I}{\int_{-\Psi}^{+\Psi} ds/I} = \frac{\int_{-\Psi}^{+\Psi} r^2 \cos\varphi\,d\varphi}{\int_{-\Psi}^{+\Psi} r\,d\varphi} = \frac{r\sin\Psi}{\Psi} \quad (12)$$

(I = moment of inertia of the cross section is supposed to be constant; y_1 is to be measured from the centre of the circle.)

N_0 is the constant thrust of the three-hinged arch. In any section i-i of the arch, the moment and forces are (the origin of the co-ordinates being point C*):

$$\left.\begin{aligned} M &= -Hy \\ N &= N_0 + H\cos\Psi \\ Q &= -H\sin\Psi \end{aligned}\right\} \quad \ldots\ldots \quad (16)$$

and the stresses at the limit of the section are:

$$\sigma = \frac{N}{d} \pm \frac{6M}{d^2}.$$

Introducing N and M, it will be found that: (Fig. 13.)

$$\left.\begin{matrix}\sigma_1\\ \sigma_2\end{matrix}\right\} = \frac{N_0}{d} + \frac{H}{d}\left\{1 \mp 6\beta^*\left(1 - \frac{\sin\Psi}{\Psi}\right)\right\}$$

$$\left.\begin{matrix}\sigma_3\\ \sigma_4\end{matrix}\right\} = \frac{N_0}{d} + \frac{H}{d}\left\{\cos\Psi \mp 6\beta^*\left(\cos\Psi - \frac{\sin\Psi}{\Psi}\right)\right\}$$

It is possible to show that in all cases

$$\sigma_3 < \sigma_2 < \sigma_1 < \sigma_4$$

σ_3 and σ_2 may be negative tensile stresses.

The French theory of " effective arches " (voûtes actives) assumes that it is always possible to draw inside any real arch AA′BB′ (Fig. 14) with the thickness d_1, a thinner circular arch AA*B*B′ with constant thickness d*, which is not submitted to any tensile stress.

It can be proved that with a value $\sigma_3^* = O$ instead of $\sigma_3 < O$ we shall always have $\sigma_4^* < \sigma_4$: the maximum compression stress in the arch will be reduced when a thinner arch is taken in order to eliminate any tensile stress.

This theory purposely ignores all that is taught in the trial load theory, particularly the yielding of the abutments which is, however, essential for stress determination. In the writer's opinion the theory of " effective arches " might be valid at the limit when the dam is fissured and being crushed by overloading (as for example at the end of a series of model tests). Furthermore, it emphasises the importance of keeping the arches at the bottom of the dam thin and rapidly gives valuable information on the safety of a dam against overloading.

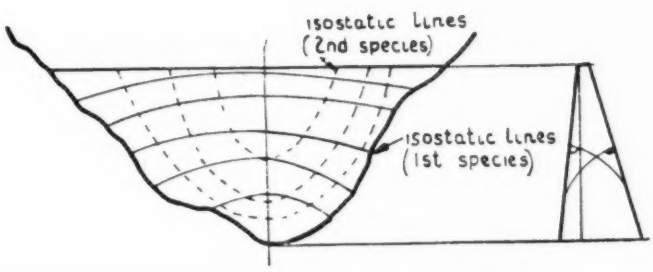


Fig. 15

B. The Diving-Arch-Theory of Coyne.

The " trial load " method, as it was first proposed at an earlier stage, assumed implicitly that the principal stresses were lying in horizontal and vertical planes, thus justifying the sub-division of the dam into a grid of vertical cantilevers and a grid of horizontal arches, the whole weight of the masonry being carried by the vertical cantilevers, the horizontal water load being divided between the two systems. Recent tests and measurements, which were carried out mostly in France, show that the lines of principal stresses do not lie on a plane, but on curved surfaces, giving the impression that part of the weight of the dam is carried by the arches (Fig. 15).

These tests and further dam measurements were the starting point of the " diving arch " theory of

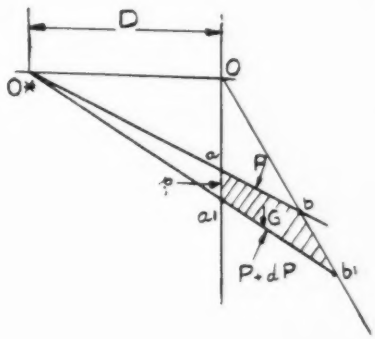


Fig. 16

A. Coyne, which is undoubtedly a valuable addition to modern arch dam theory (Figs. 16 and 17).

It is clear that the French methods implicitly and explicitly include a criticism of the trial load method, and Coyne's method in particular introduces a completely new point of view.

C. The Italian Attitude.

The Italian attitude as set out by Prof. G. Oberti, is somewhat different. He also ignores in his calculations the effect of the cantilever on the grid, but he does include the yielding of the abutments of the arches and supplements the calculation with extensive research work

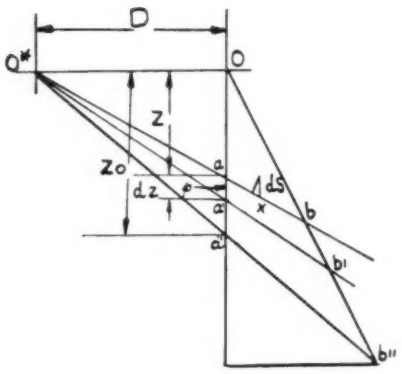


Fig. 17

on dam models. The recent Italian dams are extremely bold ones of the cupola-arch type, with double curvature and special shaped abutments with peripheric joints.

Oberti also gives some hints on the possibility of applying the theory of thin shells to cupola-arch dam calculations. In any case the Italian approach to the problems involved in this type of dam seems to be a most interesting one.

The next step is to summarise our present knowledge on arch dam measurement, to compare the results with the theory and then to extend this knowledge to the analysis of very high arch dams, with particular attention to the Italian cupola-arch dams.

Measurements on Arch Dams and Model Tests

Summarising the foregoing sections, it can be seen that several different proposals have been put forward for arch dam calculation, each of which is based on early papers§ published during the years 1913 to 1922. There is a tendency for each school of thought to reject the proposals of the others, emphasising the weakness of their methods. The writer submits that each of these methods shows different aspects of a very intricate problem, and that at the present stage of our knowledge there is no reason to reject any of the results thus attained. To further our aim, we shall now compare the theoretical results with some actual measurements.

§*The theory of Prof. M. Ritter is based on H. Ritter's paper* (1913), *the trial load method on Noetzli's paper* (1922) *and the "active arch" theory on Résal's paper* (1919).

It is generally agreed that model tests will yield the right answer to dam analysis and that the results of the tests can be compared with measurements on site.

A. SIMILARITY RELATIONS FOR DAM MODELS.

To express the similarity relations, the following notations are used :—

E_m = modulus of elasticity of dam model material

E_d = modulus of elasticity of prototype material

λ = scale ratio of prototype to model (about 1 to 50 to 1 to 100)

δ^* = specific gravity of model liquid (mercury $13.6^t/m^3$).

The relations between deflections measured on the prototype and on the model will be:

$$\rho = \frac{E_m \lambda^2}{E_d \delta^*}$$

The similarity relations between the unit deformation is:

$$\rho_1 = \frac{E_m \lambda}{E_d \delta^*}$$

The similarity relation between stresses is:

$$\rho_2 = \frac{\lambda}{\delta^*}$$

Suppose that μ is the ratio of the crushing strength of the concrete to the crushing strength of the model; the load on the model when crushing will be produced by a system of hydrostatic jacks, representing an ideal liquid with a specific density of δ_{cr}^* by which the model is supposed to be ruptured. The similarity relation in this case is:

$$\rho_{cr} = \frac{\lambda}{\delta_{cr}^*}$$

and the safety factor ε against rupture of the dam itself is:

$$\varepsilon = \frac{\mu}{\rho_{cr}}$$

B. SOME RECENT MEASUREMENTS AND THEIR INTERPRETATION.

(*a*) *American Measurements and Model Tests.* In 1922 the Committee on Arch Dam Investigation was set up in the U.S.A. for the purpose of

thoroughly studying the problem of arch dams. One of the first tasks of this Committee and, undoubtedly, the most important, was the construction of the Stevenson Creek dam, purely for experimental purposes. In close co-operation with the Committee, the Bureau of Reclamation built a concrete model of the Stevenson Creek dam on a scale one twelfth of the prototype, and a concrete model of the Gibson dam. The results of the tests on the Stevenson Creek Model were entirely satisfactory and confirmed the usefulness of test models in arch dam analysis.

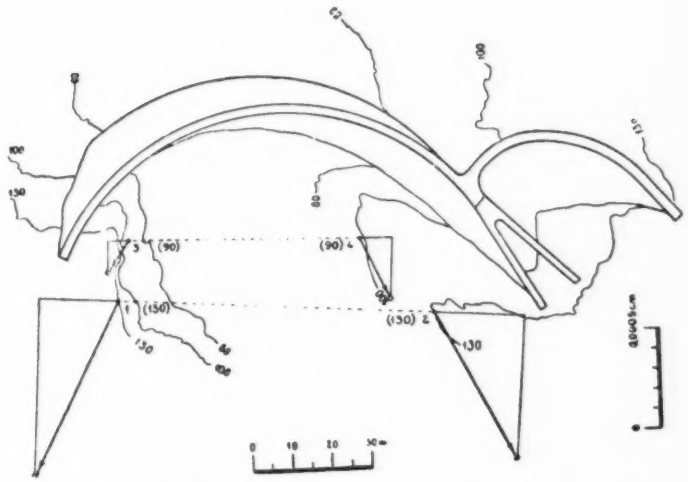


Fig. 18. *Plan of the Santa Luzia dam, Portugal*

In 1930 the Bureau of Reclamation, in order to check the results of the trial load analysis of Boulder Dam, initiated the construction and testing of models of this dam. These investigations are, undoubtedly, very important and mark an outstanding step in the development of the study of structures by the use of test models. The models were calculated by the trial load method and the agreement between computed and observed results was considered very satisfactory. In the same year, tests on a rubber model of the Calderwood dam were made at the Aluminium Research Laboratories.

The accuracy of the " trial load method " seems to be well established by the extensive research work carried out in the United States. Recent publications, particularly several reports presented to the Third Congress for Large Dams, express some doubt about the soundness of these conclusions. In particular, the French propose to abandon the trial load method as being inadequate for proper representation of the strains and stresses in an arch dam.

Reviewing all these publications, the evidence gathered in favour of abandoning the trial load method does not appear convincing. The points under discussion are chiefly the agreement between measurements on models and/or on dams and the calculations. Another point which has been raised in several reports is the direction of the principal stresses on the upstream and downstream faces, which is reported not to be in agreement with the fundamental hypothesis of the trial load method.

(*b*) *The Santa Luzia Dam, Portugal.* (Fig. 18.) Strain measurements and deflection measurements were made on the model and on the dam. There was a very sound, satisfactory agreement between the radial deflections measured on the model and those observed on the dam, whereas the deflections calculated analytically by the independent horizontal arch method, differed greatly from the above. However, it must be emphasised that the Santa Luzia Dam is asymmetrical, the right abutment being on rock, the left one on an intermediary gravity section which also supports the thrust of another smaller lateral arch dam. Owing to this gravity dam abutment, it is difficult to find a suitable analytical method of calculation for the Santa Luzia Dam. Even the trial load method would not be easy to apply.

The directions of the principal stresses, at the points for which comparison was possible, agreed well. The magnitude of the horizontal stresses observed on the dam agree much more closely with the values measured on the model than with those computed analytically. It was observed that the direction of the principal stresses, instead

of being always horizontal, is inclined downwards at some points near the abutments, particularly near the base of the dam. The measurements on the Santa Luzia Dam and the corresponding model tests represent a document of great value, but the unsymmetrical design of the abutments makes it impossible to utilise the results for comparison with other dams.

(*c*) *The upstream cofferdam of St. Etienne-Cantales, France.* This cofferdam is a thin arch dam 18 m. high, with 45 m. radius and 75 m. crest length. The measurements were carried out during the years 1936-1939 with the so-called Rabut apparatus consisting of "Invar" wires measuring radial displacements of some points on the downstream face of the dam. The great sensitivity of the thin arch to temperature variations became evident. No attempt was made to compare the measured deflections with the calculated values.

(*d*) *The Fabrèges Dam, France.* This dam was built in 1941-47. It is a thin arch dam 50 m. high, with 85 m. radius at the dam crest, 168m. crest length ; the dam thickness varies from 2.50 m. to 12.50 m. from top to bottom. Strain measurements were carried out from May 13 to May 20, 1947, the reservoir being filled during this short period. It was assumed that temperature variation effects on the dam would be negligible in such a short time. Stresses were calculated assuming a modulus of elasticity of E = 200,000 kg. per sq. cm.

There is definitely a tendency for the horizontal thrust to be thrown downwards near the abutments at the dam base.

(*e*) *The Gour-Noir Dam, France.* The Gour-Noir dam is a relatively small thin arch dam with a height of 133 ft. (40 m.), a thickness of 17 ft. 8 in. at the base and 5 ft. 7 in. at the top and a developed length of crest of 334 ft. 7 in. The radii of the intrados vary from 75 ft. at the base to 170 ft. at the top of the dam. This arch dam is nearly symmetrical.

Strain measurements were carried out during the filling of the reservoir (end of summer, 1946) and during a quick emptying in the following winter. It was noted that the deformations were identical during the filling and during the emptying of the reservoir, but strains and stresses differed from one process to the other. No attempt has been made to explain these fundamental discrepancies. Furthermore, the following facts were noted:

The annual temperature variations will deform the dam at least as much as the water pressure.

The pressures are far from being horizontal in spite of the thin wall.

The left abutment base was almost free from stresses owing to a change in the slope of the supporting rock.

The stresses are not symmetrical in spite of the geometrical symmetry of the construction.

At mid-height, towards the base of an old shaft, though well refilled and grouted, the concrete was cracked upstream by a transverse bending stress with concomitant increase of the down-stream compression.

Measurements have shown that the arch did settle during the first filling.

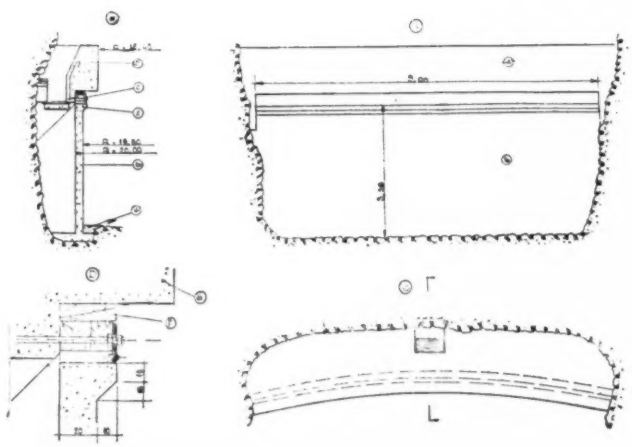


Fig. 19. *Breakdown tests on a model arch dam*

It is said in the report that measurements did not confirm the classical calculations but there is no hint as to what is meant by these calculations.

(*f*) *Measurements and Breakdown Tests of an Arch Dam, France.* An arch 3 m. high (10 ft.) and 9m. wide (30 ft.) with approximately rectangular cross section was built by the *Service Technique des Grands Barrages* in 1945, to stand progressively increasing hydrostatic heads and to be finally destroyed by breakdown tests. The arch thickness was 0.20 m. and the radius 20 m. The experiment consisted of the study of a lower slice of dam and the tests were carried out under pressures varying between 12 and 23 m. of water pressure and breakdown occurred under a pressure of 31m. of water.

Fig. 21. *The Rossens dam under construction*

The breakage (Fig. 19) started with the collapse at one abutment of the crest of the dam, which was 0.30 m. thick (1.5 times the dam thickness). This crest was too stiff and failed by crushing in compression of the concrete, leading to the opening of a vertical crack, which extended along a horizontal construction joint producing a warping of the whole panel of the arch. Numerous measurements were made of strains and displacements. Stresses were calculated on the basis of E = 420,000 kg. per cu. m., resulting from samples taken from the arch itself. These results were compared with theoretical results. The radial displacements were measured at four different levels, at four points each time (16 points) and it was found that the ratio:

$$r = \frac{\text{measured displacement}}{\text{calculated displacement}}$$

is a function of the depth h. (Fig. 20). The calculations were made taking into consideration independent horizontal arches built in at both ends.

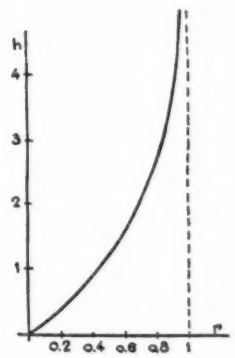


Fig. 20.
Ratio of measured displacement to calculated displacement plotted against depth for a model arch dam

(*g*) *Provisional Conclusions on the Tests and Measurements.* Some provisional conclusions can be drawn from this series of measurements and tests.

The isostatic lines along the dam faces are not horizontal as commonly assumed in the "trial load method," but curved and inclined towards the arch abutments, being nearly perpendicular to the rock surface at their end. This has been observed also on the model of Santa Luzia Dam and on some Italian dams (see later) as on many French dams. There is therefore, some justification for the proposal of M. A. Coyne to consider not horizontal "arch slices" but "inclined arch slices."

The fissures and crushing produced when overloading an arch dam, just before final breakdown, are very similar to those assumed in the "effective arch" calculation. This method should, therefore, be the right one for investigations on the final stage

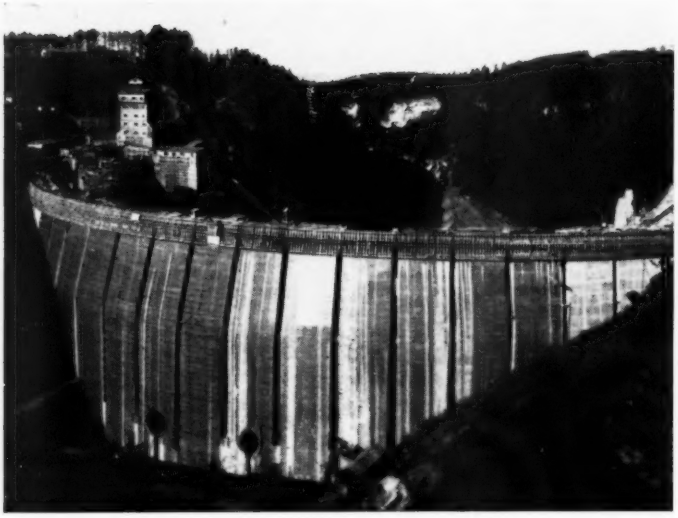


Fig. 22. Upstream view of the Rossens dam showing final lift being placed in position and open construction joints

of the dam before final collapse. Research work on these lines would be welcome. There is no real justification to be found in favour of the "effective arch theory," so long as the dam is not fissured or damaged by cracks. Plastic deformations are ignored in this theory.

The case put forward in several papers, and even in text books, against the "trial load method" is certainly strong in theory, but has not been proved in fact. None of the dams tested in recent years which have been described in this chapter has been analysed by the "trial load method." The calculated stresses were on the basis of sometimes rather rough assumptions or on the basis of the built-in arch theory, ignoring the yielding of the arch abutments. No comparison on this restricted basis can be conclusive. It has, on the contrary, been proved by all tests that cantilever action and abutment yielding should not be neglected.

Very High Thin Arch Dams and Cupola Arch Dams

A. Some Recently Built Very High Arch Dams

It is considered that a special section should be devoted to some extremely high, thin arch dams recently built or at present under construction or consideration. These dams are probably amongst the boldest constructions ever undertaken and compare with unique achievements like the Plougastel bridge in Brittany, France, and the large suspension bridges in New York and San Francisco.

The Rossens Dam (Fig. 21) is part of the Thusy Hauterive power station, Switzerland, the capacity of which has been increased from 24,000 h.p. to 80,000 h.p. through the construction of the dam—an increase of output from 50 to 230 million kWh. This dam was designed by M. H. Gicot and had been calculated according to M. Ritter's method. The total dam height is 83 metres, the crest length 320 metres; the radii of the arches vary from 105 metres at the dam base (level 600 m. O.D.) to 165 metres at the crest (level 677 m. O.D.). The arches are circles with constant thickness. The rock, in which the dam is built, is sandstone. The reservoir (100 million cubic metres capacity) is connected to the power house by a tunnel 5.9 km. long and 5 m. dia. The gross head varies from 69 to 100 m., the maximum discharge being 75 cu. m. per sec.

The main data concerning recent French dams and dams of French design are summarised in the following table:—

Dam	Height	Length of crest		Thickness of the crest	Thickness of dam base	Volume
	m.	m.	L/H	m.	m.	m^3
Sta. Luzia	75	115	1.5	2.5	12	
Castelo do Bode ..	115	350	3.0	6.0	52	
Aigle ..	90	290	3.2	5.50	45.50	240,000
St. Etienne—Cantalès ..	70	270	3.9	5.0	35	130,000
Marèges ..	90	247	2.7	3	19	185,000
Castillon ..	101.40	200	2.0	4	16.80	125,000

Very little is known about the two dams recently designed by M. A. Coyne for Tignes, Val d'Isère,

France, and Cabril on the Rio Zezere in Portugal. The Tignes dam will be 180 m. high and will provide storage for 233 million cu. m. of water at a level of 1,790 m. O.D. The water from Tignes reservoir will operate three Francis turbines (each 43,800 h.p.) at the Les Breviaires power station and four Pelton wheels (each 101,500 h.p.) at Malgovert station (altogether 537,400 h.p.). The Cabril dam is 125 m. high and part of the great Portuguese Rio Zezere Valley scheme. Apart from the increased height and slender profile, these dams seem to be of the type developed by M. Coyne for l'Aigle in France and Castelo do Bode in Portugal. More details are available on some outstanding Italian dams, such as:—

Fig. 23. *Downstream view of L'Aigle dam and power house*

The Osiglia Cupola Arch Dam, 75 m. high and with an arch chord length of 220 m. The volume of the dam is 75,000 cu. m.

The Pieve di Cadore Dam on the River Piave was finished in November, 1949. This dam is 110 m. high, with varying thickness, from 5m. at the crest to 35.81 m. at the base. The length of the dam crest is 410 m. The volume of the dam is 400,000 cu. m.

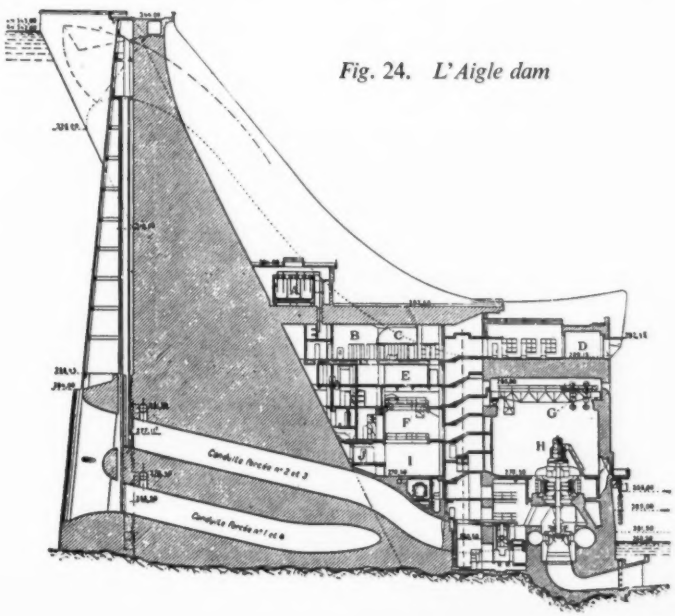


Fig. 24. *L'Aigle dam*

The Lumiei Dam (1943-1947) on the River Lumiei, tributary of the Tagliamento. The height of this cupola arch dam is 136.15 m., the width varies from 3.15 m. at the top to 13.68 m. at the base of the dam, the width of the dam foundations being 15.60 m. (The length of the dam crest chord is 130 m.). This dam is very remarkable because of its symmetry and smooth development of the slopes of the side abutments, resulting from increased excavations on both sides of the valley. The volume of the concrete used for the dam is 100,000 cu. m.

The Vajont Dam Project with a height of 202 m., width of 2.90 m. to 16.00 m. and foundation width of 17.50 m. is, for the time being, the highest thin arch dam under

consideration. It belongs to the same system as the Pieve di Cadore dam.

The Pieve di Cadore Dam, the Lumiei Dam and the Vajont Dam all belong to the Società Adriatica di Elettricita. *The Sta. Giustina Dam* on the River Noce (Italian Alps, north of the lake of Garda) is being built by the Società Edison. The height of this reinforced concrete arch dam is 152.50 m. (500 ft.) with a thickness of 3.50 m. at the crest and 16.50 m. at the dam base. It will be one of the highest arch dams in the world being second only to the Tignes Dam and Vajont Dam in Europe and to the Ross Dam in the U.S.A. (170 m. high). The total volume of the dam is 120,000 cu. m. The foundations are on Dolomite limestone, with extensive grouting.

B. Italian Method of Cupola-Arch Dam Analysis.

All these Italian dams are of cupola-arch dam type and are remarkable for their very thin arches at the dam base. Theoretical studies have been developed on this type of dam but they assume that all the centres of curvature of the arches are on the same vertical line. This is certainly not the case with cupola-arch dams, so that, for the time being, theoretical research work on the lines of the theory of thin arched shells has been abandoned. According to Oberti the effect of the second curvature is to reduce the tensile stresses in the vertical cantilevers.

It was decided that the dams would be calculated taking into account only the system of arches (and

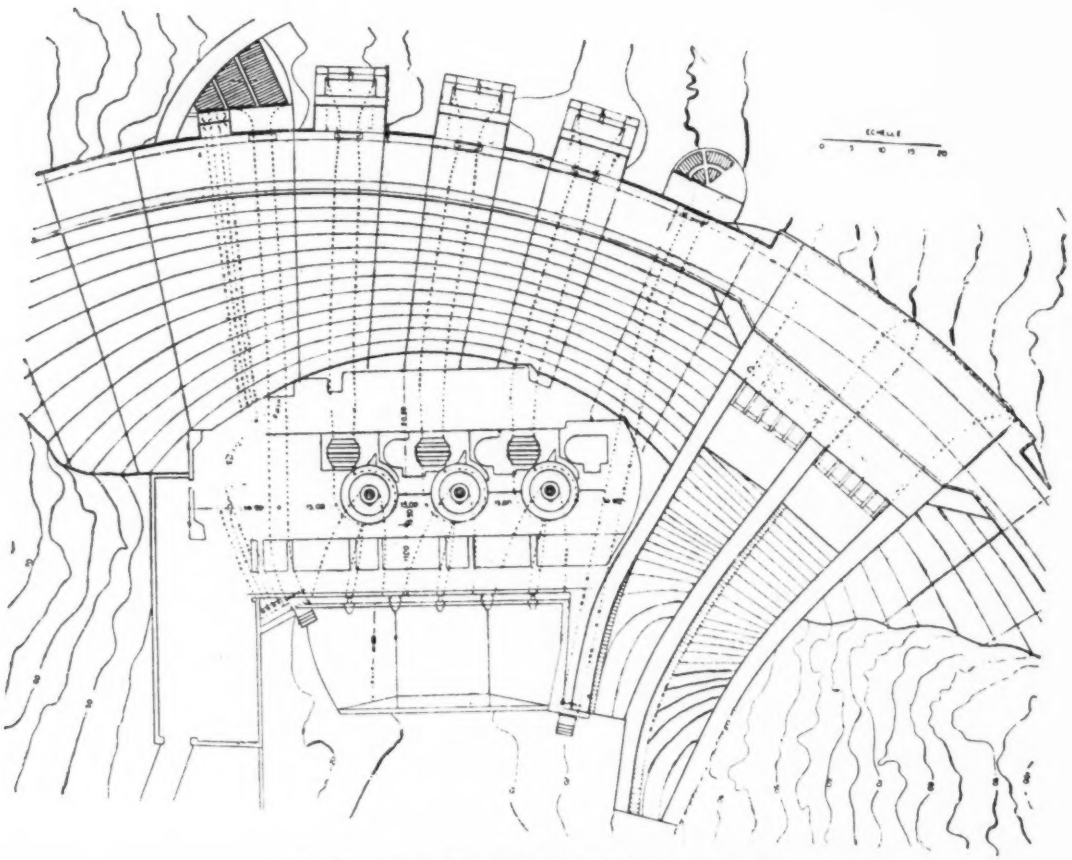
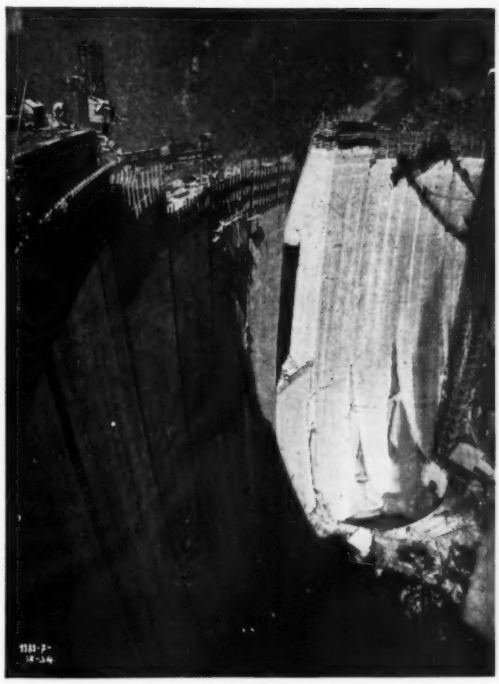
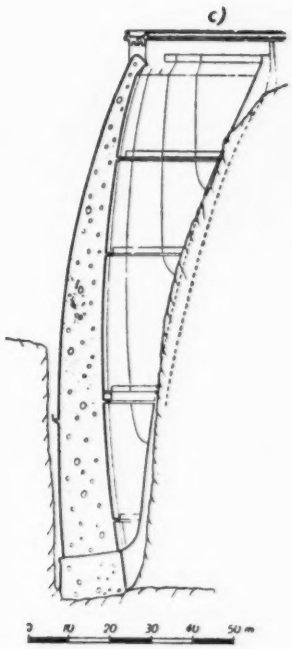
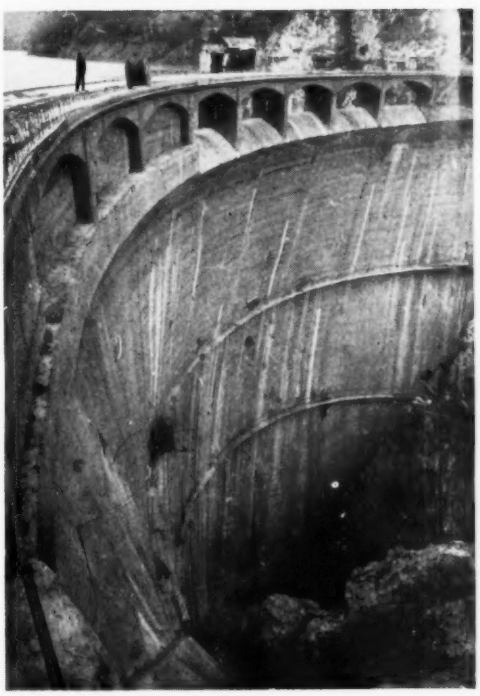


Fig. 25. *Plan of the Castelo do Bode dam, Portugal*







Top left: *Fig.* 26. *The Mareges dam under construction*

Top right: *Fig.* 27. *Cross section through the Lumiei dam*

Bottom left: *Fig.* 28. *The Lumiei dam*

not the cantilevers). The arches have a variable thickness which increases slightly against the abutments, and were calculated as built-in arches ; the lower arches were calculated as thick tubes, taking into account the non-linear stress distribution in thick tube walls. All the calculations were systematically checked by extensive model tests in the Laboratorio prove costruzione e modelli del Politechnico di Milano. (A new laboratory for arch dam testing on models is being developed at Bergamo.) A point which has to be stressed here is the very high degree of skill shown by the Italians in these researches, which the writer considers of fundamental importance in the case of very high thin arch dams of the type under discussion. The

Fig. 29. *The St. Etienne dam*

general tests on dam models were supplemented by special research work on photo-elastic models and tests on different types of abutments.

C. Provisional Conclusions on Italian Practice

Provisional conclusions can be drawn from Italian practice as follows:

(i) The first is undoubtedly that model tests should proceed simultaneously with the designing of the dam and precede the final design. Model tests, as used by the Italian experts, are no mere checking of results obtained by calculation, but are integral parts of the dam analysis.

(ii) The second is that the Italian designs are the logical conclusion of the French theory of "effective arches." This French theory shows how the tensile stresses are suppressed and compression stresses reduced when thin arches are used instead of thick ones—at least when the abutments are presumed to be non-elastic. The Italian design of cupola-arch dams takes full advantage of these results and leads to dams which are extraordinarily thin at the base.

(iii) G. Oberti stresses the importance of the symmetry of arch dams and C. Semenza did not hesitate to excavate 63,000 cu. m. of rock in order to achieve the perfect symmetry of the Lumiei dam.

(iv) In his articles, Oberti emphasises the advantages of the double curvature of the cupola-arch dam design. This double curvature reduces tensile stresses in the cantilever elements. The tests prove the importance of the cantilever action in cupola dams. Maximum stresses in the arches of the Lumiei Dam do not exceed 60 kg. per sq. cm. (850 lb. per sq. in.).

(v) Oberti and Semenza have introduced an entirely new design for joints and are in favour of a peripheral joint along the abutments.

(vi) When discussing the results of the breakdown test on a French dam, it was pointed out that, at this stage of overstressing, the dam no longer works as a three-dimensional shell (as assumed in the trial load method) but as a system of independent arches as assumed in the French method of "effective arches." Oberti

Fig. 30. *The Osiglia dam*

comes to the same conclusions. Furthermore, it is observed that the cracks and fissures in a dam on the point of failure are horizontal and vertical, there being no evidence of curved lines of maximum stress on which the theory of Coyne is based. At the first stage of normal loading and elastic deformation, the effect of the cantilevers is certainly more important than the inclination of the maximum stresses; at the last stage of final failure of the dam, when the cantilever action is almost entirely suppressed, the lines of rupture are horizontal or vertical cracks.

(vii) Finally, Italian experience shows that an extremely high factor of safety obtains in thin arch dams. It has been calculated that the safety factor of the Lumiei Dam, based on the breakdown tests of the model and the crushing strengths of mortar and of concrete, is about 12. No gravity dam nor buttress dam of any type can ever attain such safety factors.

Fig. 31. *The Osiglia dam*

Some of the provisional results mentioned here are of sufficient importance to be dealt with again later.

Comments on the Data Actually Available on Large European Arch Dams and Recommendations

On an intricate and difficult subject such as arch dam analysis and design, special theories on dam analysis or material gained by odd measurements or observations may seem of little value if considered by themselves. The value of these facts and theories becomes obvious when compared with other facts and theories, and considered in connection with all the other material it is possible to gather on the same subject. No decision to reject or accept a new idea should be taken, unless comparison with the whole subject matter has been made. This emphasises the importance of collecting all available facts and theories about modern arch dams, examining, and commenting on them.

Some provisional conclusions have already been reached on arch dam measurements and on the Italian practice, and these should be reviewed again before making more general comments.

A. Proposal for a Method of Reference for Arch Dam Calculation.

Despite the considerable pioneer work done by American engineers, the " trial load method " has very seldom been used in Europe. This can probably be explained by the fact that many engineers would prefer a straightforward method of calculation, rather than one of trial and error, in spite of the undoubted advantages to be gained by using the American method.

Another possible objection is the fact that " trial load methods " involve an amount of specialised calculation for which few engineering concerns or public works offices could find the staff and afford the cost. The writer does not think there is a strong case against the accuracy of the trial load method, as sometimes has been argued. Possible discrepancies between theory and measurements can be explained otherwise, mainly by the fact of insufficient theoretical research, of neglecting to include in the calculations the yielding of the arch abutments, or by asymmetry of the actual dams which were calculated as symmetrical ones.

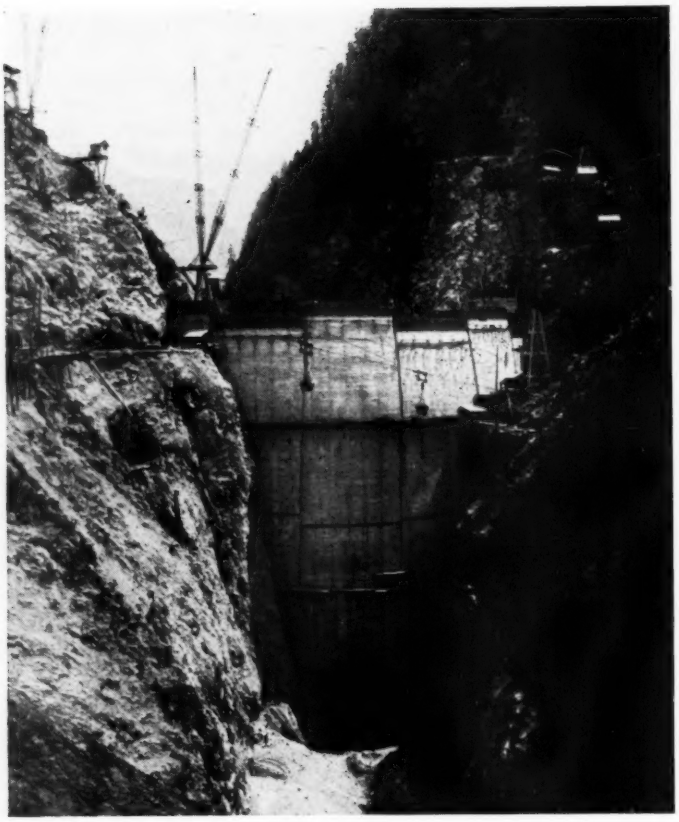


Fig. 32. *The Lumiei dam under construction*

It is possible that the fact that the isostatic lines on the dam faces are not horizontal straight lines but are curved and inclined near the abutments may lead to some modification of the trial load method. This is a fundamental aspect which should be investigated thoroughly. There are several cases where large arch dams have been calculated by a "simplified trial load method," taking into account only the radial deflections in the dam axis (or near the dam axis in one radial plane only) and, therefore, considering the deflections of one cantilever only. Furthermore, the calculations referred to did not include the effect of the displacement and yielding of the abutments. The effect of the elasticity or plasticity of the abutments is an integral part of the calculation of an arch dam and cannot be ignored. It is more important to include this effect than to extend the calculation to several cantilevers.

There is a strong case for the proposal of M. Ritter's method as a sound alternative to a too simplified trial load method, which appears to be the present tendency. Ritter's method is a straightforward one which will appeal to many an engineer's mind, because of the exclusion of any "trial" approach. The method is far shorter than any "simplified" trial load calculation and conveniently includes the yielding of the abutments. Ritter's theory can be used for the calculation of temperature stresses or similar cases and can be extended to a more complete calculation, including several cantilevers or asymmetrical shapes of the arch dams.

B. Rock Elasticity and Plasticity. Improved Model Test Method.

The importance of including the yielding of the abutments in the calculation of the arches as well as in the calculation of the cantilevers has already been emphasised. Yielding of the foundations was introduced years ago in the calculation of bridges. Pressure tunnel technique also has taken into account the elastic and plastic deformations of the rock since the tests on the Amsteg pressure tunnel after the failure of the Ritom tunnel in 1924. Exact measurements of the modulus of elasticity of the rock on the foundation site of any arch dam may now be considered as essential. Particulars are given below of experiments which have been carried out in connection with arch dam construction and are accompanied by the writer's comments.

The tests carried out by Mr. O. Frey-Baer for the Lucendro tunnel were most valuable for the construction of the nearby dam of Lucendro and the results obtained there have a more general value. Measurements were carried out on several stretches of the tunnel in order to test different types of gneiss. The hydrostatic water pressure

used for the tests was about 10 kg. per sq. cm. Rough and lined tunnel sections were tested. The method of testing consisted of measuring the length variations of two diameters at angles of 90 deg. The measurement apparatus was of the Gerber type, built by Dr. A. Huggenberger. It was found that in some cases important plastic deformations of the rock occur. After repeating the tests several times, the rock was shown to become elastic. As an example, there was found in Paragneis an overall modulus (including plastic deformations, called " Verformungsmodul " by Frey-Baer):

V_f = 120,000 kg. per sq. cm.

the elastic modulus being:

E = 390,000 kg. per sq. cm.

In Fibliagneis, it was found that:

E=63 to 79,000 kg. per sq. cm.

A concrete lined section of the tunnel, with steel reinforcement bars, proved to have a:

V_f=150,000 to 230,000 kg. per sq. cm.

20 to 28 per cent. being plastic deformation.

The technique used by Oberti in fissured limestone is very similar to that used by Frey-Baer and the results are on the same lines.

Mr. H. Gicot used a completely different technique for the Rossens Dam. The rock there is sandstone in deep or shallow layers. The modulus of elasticity was measured there by applying a concentrated load P on a circular plate (diameter d). The formula of Vogt gives:

$$E = \frac{m^2-1}{m^2} \frac{P}{d\delta}$$

($m = \frac{1}{\nu}$ and δ = the deformation under the circular plate.

ν=coefficient of Poisson).

The device used by Gicot was built so that loads could be applied both in the vertical and in the horizontal directions. Tests were carried out at several points along the abutments of the future dam. It was found that E varies with the direction of the strata, E being smaller in a direction perpendicular to the strata than in a direction parallel to the strata.

Figures measured in both directions are:

	normal to strata kg. per sq. cm.	parallel to strata kg. per sq. cm.
Rossens	22,000	50,000
Urseren granite ..	250,000	400,000
Urseren Schists ..	80,000	280,000

It has been observed at the Gour Noir Dam that strain distribution may be asymmetric in spite of the symmetric shape of the valley. The

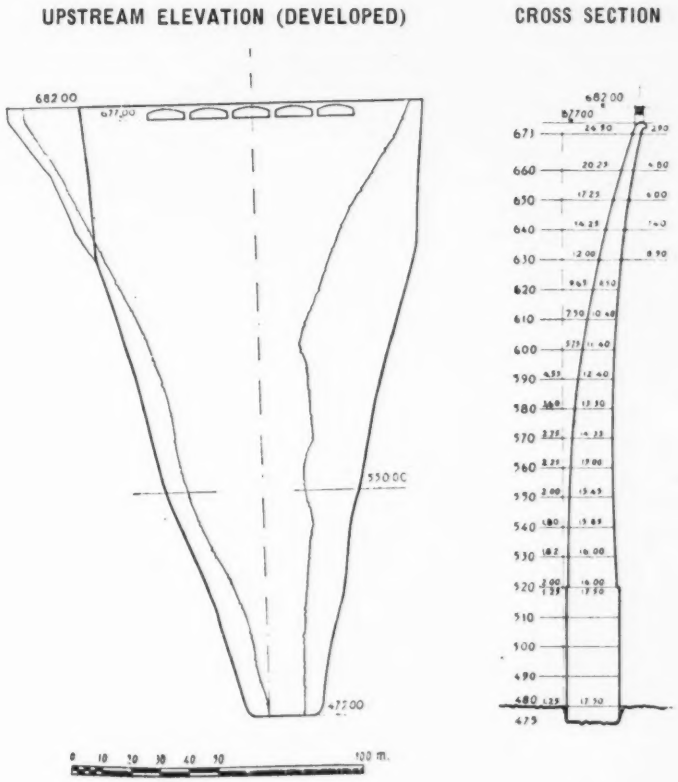


Fig. 33. Upstream elevation of, and cross section through, the Vajont dam

most likely explanation for this fact is that geometrical symmetry can be disturbed by geological asymmetry, due to inclination of geological strata, and this strengthens Mr. Gicot's view that the measurement of E should be carried out at different levels along the dam foundations in order to check the variations of E. Mr. Caquot has observed at the Sautet and Guerlèdan dams that fixed points downstream of the dam, which were used for measuring the dam deflection according to the method of Lang, had moved. Similar observations were made at Nepean Dam by S. T. Farnsworth. Both authors explain the measured distortion by water penetrating the fissures of the rock. Foundation deformations have also been measured by Messrs. M. Rocha and J. Laginha Serafim on the model of Santa Luzia Dam. These deformations cannot be explained by water penetrating into the model fissures. They are due to the elastic yielding of the dam foundations in the model and are greater at a higher level than at the bottom of the valley. It may well be wondered if this is not the right explanation also for the rock distortions measured in France and Australia, rather than water penetrating through fissures in the rock. Mr. H. Gicot has stated that he tried to analyse the effect of the vertical water load behind the Rossens dam.

In all cases, elastic and plastic deformations of the valley have to be investigated and taken into account in future dam analysis and tests. For dam calculations the ratio E rock/E concrete is essential. The exact value of E concrete should also be checked. Values assumed in calculations on the basis of laboratory tests vary between 200,000 kg. per sq. cm. and 400,000 kg. per sq. cm. In the writer's view the value of E concrete should be measured on the dam itself and possible variations of E concrete should be checked over a period of years. This could be done by artificial shock waves produced on the dam and in particular by water hammer waves produced in the drainage system or inspection shafts of the dam after filling with water.

Modern dam designers are increasingly aware of the paramount importance of the elasticity of the dam abutments. The next step is, obviously, to introduce the right ratio E concrete/E rock in model tests. This has already been done by Messrs. M. Rocha and J. Laginha Serafim for the Santa Luzia Dam model and it is understood that Italian and Austrian engineers will do the same in their future tests. The focus of interest which, several years ago, was concentrated on methods of dam analysis, is now gradually shifting to the boundary conditions on which the ultimate safety factor of the dam depends.

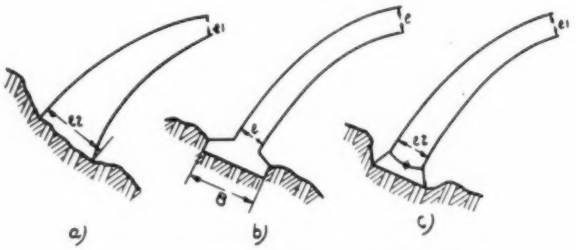


Fig. 35. *Three different types of abutment*

C. THE BOUNDARY CONDITIONS. SHAPE OF ABUTMENTS AND DAM CREST.

If the abutments are elastically yielding and are to be considered as part of the dam project, then transmission of forces and moments from the arches and cantilevers to the rock is essential, and the shape to be given to the dam abutment has to be investigated. In Fig. 35 three different designs for arch abutments are shown. In (*a*), the arch is gradually broadened from the dam axis towards the abutments and forces are transmitted to the rock over a large area. This type of abutment can be found in North American dams and is becoming characteristic of the French dams designed by Mr. A. Coyne. In (*b*) is shown the solution arrived at by Messrs. Ritter and H. Gicot for the Rossens dam. The arch remains slender, with constant thickness over its whole length. Forces and moments are transmitted from the arch to the rock through a saddle which may be asymmetrical with respect to the arch centre line, and is specially analysed in order to transmit equally distributed stresses to the rock, both when the reservoir is filled and when it is empty.

The third figure (*c*) is the result arrived at by the Italian designers. The arch remains slender

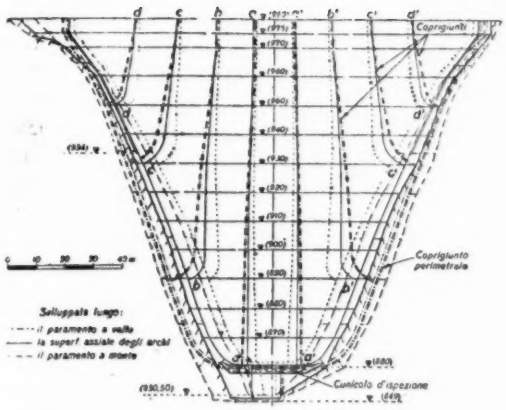


Fig. 36. *Peripheral joint at the Lumiei dam*

with a gradual increase in thickness. The forces and moments are transmitted to the rock through saddles as in (*b*) but a peripheral joint has been introduced between arch (or cantilever) and saddle. In Fig. 36 the peripheral joint in the Lumiei Dam is illustrated. This was protected against water percolation by a sheet of copper and was grouted at the end of the construction of the dam. Provision has been made for repeating the grouting if necessary. No tensile stresses can be transmitted through such a joint.

Extensive model studies have been made on concrete models of the abutments of the Lumiei Dam. The inclination of the principal stresses near the abutments, which is so apparent in the French type of dam (including the Santa Luzia Dam which was designed by Mr. Coyne), is negligible in the Italian type of dam and it may be wondered to what extent this is due to the shape of the abutments. The dam crest is usually designed to suit particular problems, such as water spilling over the crest or to accommodate a road. The dam crest may be thickened for water spilling over the top of the dam along its whole length or, on the contrary, a deep channel may be cut through the crest for a proper spillway (*e.g.*, Castelo do Bode dam).

Very little study has been made of this important problem. R. Bourriot emphasises that the breakdown of the dam he tested began with the crushing of its thick crest. At the very moment when the first signs of collapse were noticeable, the compression stresses in the concrete were about 300 kg. per sq. cm., that is, far less than the crushing strength of the concrete, which was proved to be about 500 kg. per sq. cm. Bourriot suggests, therefore, that the breakdown of the dam would have been postponed, if the reinforcement of the dam crest could have been avoided.

Similar difficulties were encountered during the study of Rossens dam, where it was found that a thick crest arch provoked tensile stresses in the cantilevers. In the first project submitted by H. Gicot, it was proposed to cut the upper arches by vertical permanent joints, in order to avoid the detrimental action of a stiff arch at the top of the dam. Later on, the design of the dam was altered and the upper part made more slender.

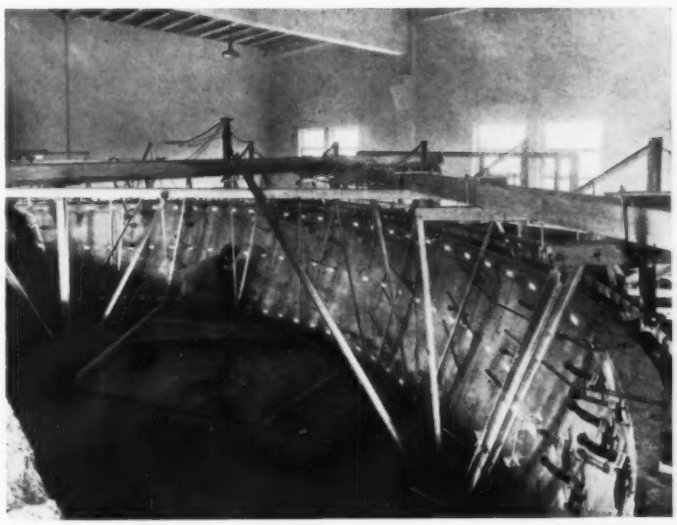


Fig. 37. *Italian model tests*

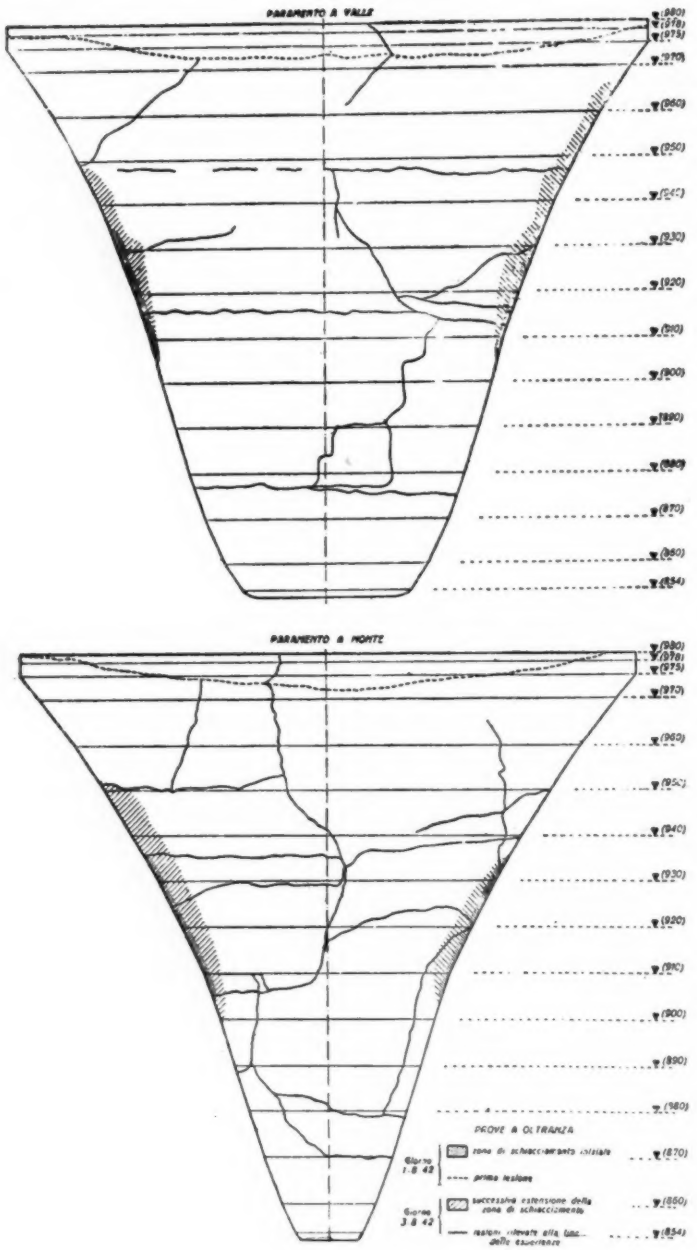


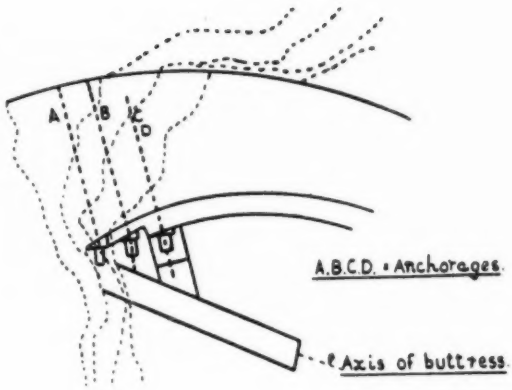
Fig. 38. Cracks obtained during model tests on the Lumiei dam

D. Ultimate Crushing Strength of an Arch Dam.

It has been stated that, according to R. Bourriot, the dam he tested for breakdown would have collapsed (without the unfavourable crest reinforcement), when stress reached about 500 kg. per sq. cm.; this is eight to ten times the calculated stresses in an arch dam of conventional design. G. Oberti calculates that the safety factor of the Lumiei Dam is about 12, based on a comparison of the crushing strengths of concrete and plaster of Paris.

No gravity dam and no buttress dam whatsoever can claim to approach such a high safety factor. Accidents on existing dams of this type have proved that the margin of safety is sometimes very much reduced. The tests and measurements carried out by G. Oberti seem to prove that an arch dam, even a slender and bold cupola arch dam of the Italian type, will not collapse by over-stressing of the central part of the shell. Any breakdown will begin somewhere at the dam boundary, most probably at the rock abutments.

These findings of recent dam analysis justify the modern trend of paying the utmost attention to the dam abutments, to the plasticity and elasticity of the rock, and to the design of the crest. The points where crushing begins must be examined and reinforced or reduced in thickness. In future the problem of consolidation of the rock near the dam abutments will be given the utmost attention. We know that the modulus of elasticity of the rock can be increased by using

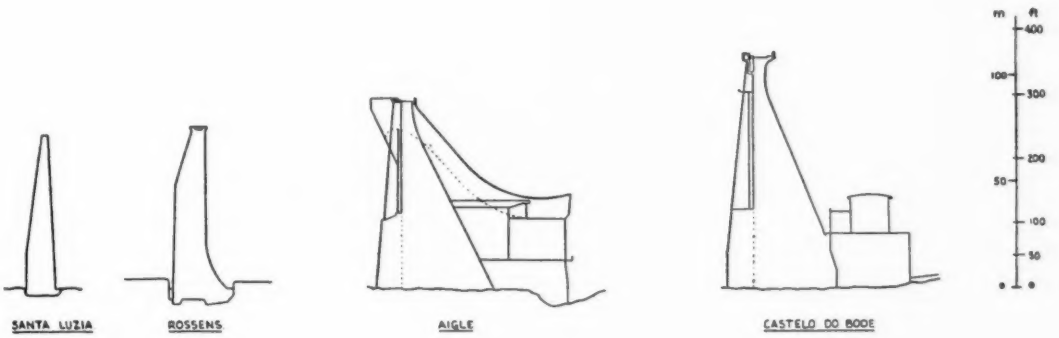


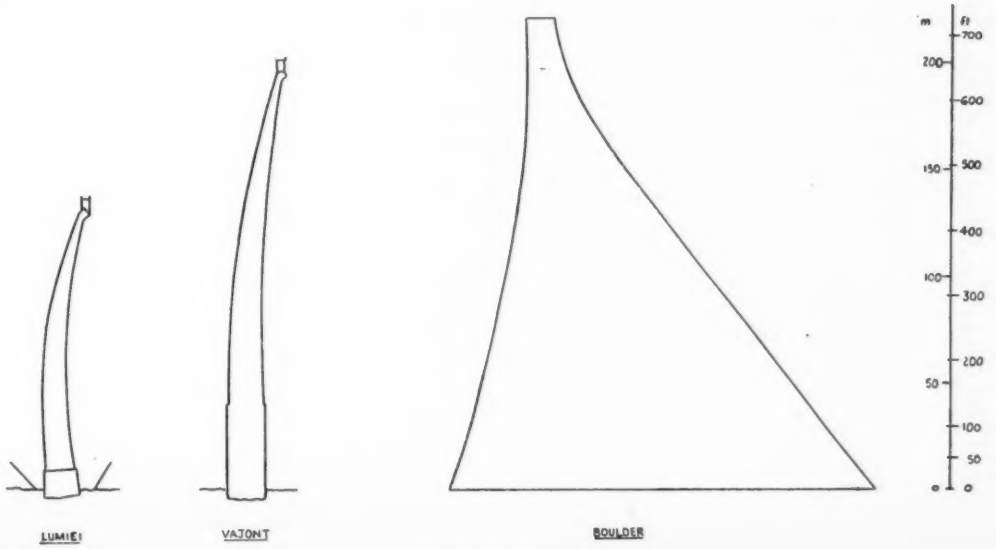
cement grouting and can expect also that cement grouting will increase the average strength of fissured rock.

It is worth while mentioning here the extremely bold repair work carried out at Castillon Dam by Mr. A. Coyne, where a fissure in the rock at the right abutment was strengthened by three cables anchored in the rock and suitably pre-stressed (Fig. 39). At the Rossens Dam a similar

Fig. 39 (*left*). *Rock anchorages at the Castillon dam*

Fig. 40 (*below*). *A comparison of the cross sections of some large arch dams*





fissure at the right abutment (on sandstone) was closed by filling with concrete and grouting.

Conclusions

The first impression gained in perusing the most recent publications on arch dam design and analysis is that there are several schools of dam designers who do not agree on fundamental questions. It has been pointed out where American, French and Italian experts differ. When analysing all the most recent studies concerned with arch dams and considering them as a whole, it appears that the gap between the respective theories is less important than our lack of knowledge on fundamental points.

There is still a long way to go in the field of arch dam design and analysis, and there is a basic need for more co-ordinated research work. Such a co-ordinated programme of research should be submitted to the next Congress on Large Dams. Among the problems which the writer urges should be investigated are further research on the lines of the method of M. Ritter. No systematic research work has ever been done on the influence of the valley shape (U- and V-shapes) on the stresses in the dam*. Dams in asymmetrical valleys need to be investigated on models. The results of the Austrian tests on the Hierzmann Dam are looked forward to with great interest. Problems concerning the boundary conditions should be investigated further (problems of shape, strength of the abutments and dam crest, geology of the valley).

Some investigations have been made regarding the modulus of elasticity of the rock. The modulus of elasticity of the concrete in the dam should be tested, particularly with respect to possible variations of this modulus with the age of the concrete. The method of shock waves could be used for this purpose. The research work begun in Italy on the influence of the construction method (grouting of the joints, inspection galleries, vertical shafts, etc.) should be continued. Finally, the greatest importance attaches to the breakdown conditions of model arch dams and research should be extended to varying boundary conditions (varying modulus of elasticity and strength of the abutments).

Strains and stresses produced by temperature variations or equivalent phenomena as, for example, shrinkage of concrete, have not been dealt with here, but it is well known that these can be even more important than stresses due to hydrostatic loads alone.

BIBLIOGRAPHY

A full bibliography has been given in the original article published in the *Civil Engineering and Public Works Review* from April to September, 1950.

* *According to E. Bosshard, the calculation of very high dams in V-shaped valleys should depart from the standard "trial load method," the trial loads on the vertical cantilevers being replaced by moments on a system of vertical beams. The trial load method is proposed for wide U-shaped valleys only.*

The Electrification of Worsted Mills in the Midlands

By J. C. EDMED, D.F.H., A.M.I.E.E., Textile Division, and J. E. COLE, Manager, Leicester Office

Although Yorkshire is regarded as the main centre of the woollen and worsted industry, there are still a number of mills located in the Midlands. It is interesting to note that about the middle of the fourteenth century, following the Black Death, the Leicestershire farmers were forced to turn from producing corn to rearing sheep, due to the shortage of labour. The wool produced at that time was exported to Calais for processing on the Continent. There is evidence, however, that woollen goods were in fact manufactured in this part of England during the fifteenth century.

The great Midland plain has often been regarded as one of the best grazing areas for sheep, and before the Industrial Revolution the sheep rearing industry provided a livelihood there for many people. Before the building of factories, the work of processing the wool and weaving was carried out by families in their own homes, and such novels as George Eliot's " Silas Marner " give some idea of the old cottage weavers in the Midlands. By the end of the eighteenth century many factories had been built and, with the availability of coal, the hand machines used by the cottage spinners and weavers were replaced by larger machines driven by steam engines through long lineshafts.

The purpose of this article is to illustrate some of the problems involved in the conversion of the old steam driven mills to electric drives, and incidentally to describe processes in the manufacture of worsted yarn. The recent electrification of Messrs. Fielding & Johnson's worsted spinning mills at Leicester and Nuneaton is taken as a good example of this kind of conversion, but before describing the installations at these factories some reference to the history of the Fielding & Johnson organisation is relevant.

In common with most textile concerns, this worsted spinning firm started from small beginnings and was founded in the year 1720 by a Thomas Johnson, Esquire, of Leicester. Even in 1720 Leicester was a prosperous town and the centre of a rich agricultural county. Mills driven by steam engines were established towards the close of the eighteenth century when cheap labour was available through the decline in the fortunes of the agricultural workers and there was a cheap and abundant supply of coal from the expanding Midland coalfields.

The existing Bond Street Mills of Fielding and Johnson at Leicester were built in 1830. The mills at this time contained hand operated machinery including such machines as the famous ' spinning jenny.' As the industrial era, which owes its birth to the steam engine, came into being, an engine of about 15 h.p. was installed. Thus, unlike many of the Yorkshire mills which made the transition from hand to steam operation through the intermediate stage of water-wheel drives, the Bond Street Mills were converted directly to steam power. It was not until 1861 that the power requirements of the mills warranted the use of a bigger steam engine. In that year Fielding and Johnson erected a twin beam engine of 80 h.p., followed in 1877 by a single beam engine of 40 h.p., both supplied by John Petrie of Rochdale. These engines were compounded about 1896 and were capable of an output of 750 h.p. following the installation of new boilers. They ran continuously until 1947 and are indeed a credit to the engineers of the nineteenth century.

Fig. 1 shows the twin beam engine used for driving the West Mill at Bond Street. The gear wheel adjacent to the flywheel drove a secondary

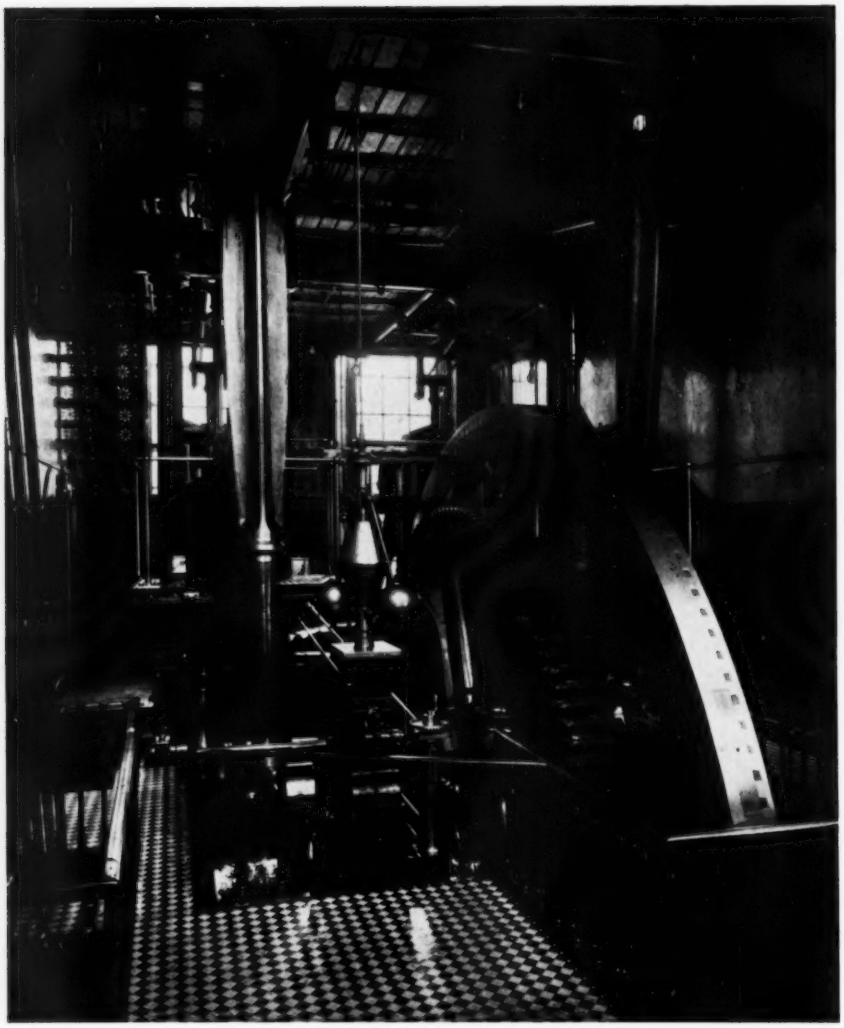


Fig. 1.
Double-beam steam engine in the Bond Street Mills of Messrs. Fielding and Johnson Ltd., Leicester

shaft which in turn drove the main vertical shaft and thence, via bevel gear, the lineshafting for each floor. A section of the vertical shaft with bevel gear drive for one of the floors is shown in Fig. 2.

Following the steam to electric conversion of their Bond Street mills, Fielding & Johnson, Ltd., decided to proceed with the electrification of the Anker Mill, Nuneaton, on similar lines, and this has now been completed. The Anker Mill was built in 1861 by local enterprise to relieve unemployment, and was taken over by Fielding & Johnson in 1886 when it was re-equipped for worsted spinning. Prior to the change-over to electric drives, the mill was driven by a twin beam engine capable of developing 750 h.p. after compounding. It is interesting to note that this engine was in operation for 350,000 working hours. The electrification of the Abbey Mill, Leicester, is now proceeding and involves a D.C./A.C. change-over from group to individual motor drives.

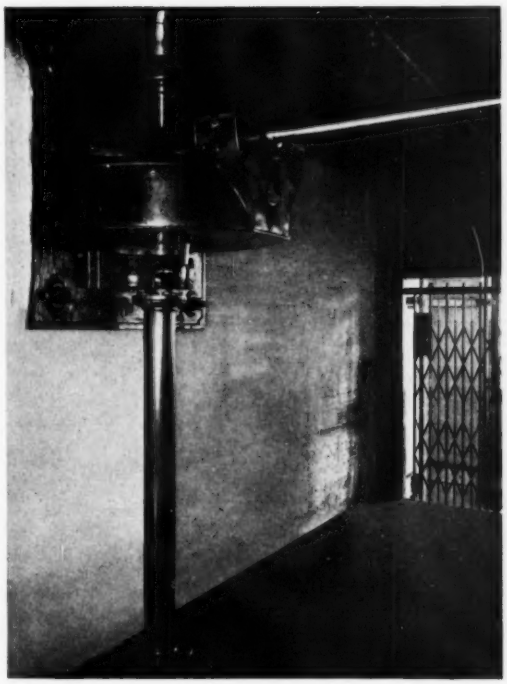


Fig. 2.—Section of vertical shaft, with bevel gear drive for one floor, at West Mill, Bond Street

Processes in Worsted Mills

Some explanation of the processes in worsted mills may be desirable in order to appreciate the electrical driving problems involved. The majority of worsted spinning mills generally process the material from the bale or 'fleece' stage to the spinning of the actual yarn, the sequence being :—

Raw wool—wool sorting—scouring—carding—backwashing—combing—drawing and roving—spinning.

The raw wool or fleece which is sent to the mills in bale form is a compressed mass of individual fleeces which contain many impurities in the form of organic greases and vegetable matter. Before processing the wool it is necessary to classify the fleeces. A fleece contains many fibres of varying quality, the coarse fibres being located on the underside of sheep whilst the finest fibres grow on the sheep's back. Normally, the fleeces are divided into their respective grades of varying quality. This 'wool sorting' is carried out by hand at the factory. After sorting, the fibres have to be opened, and to do this they are fed into a Willow machine which consists of a spiked drum and feeder lattice. Before the wool can be processed further it has to be washed to remove the organic greases and oils. All these impurities must be removed for obvious reasons before the wool is ready for combing and drawing.

It is the usual practice to pass the wool through a series of three or four 'wash bowls' with a hopper feed to remove the bulk of the foreign matter. A wash bowl is a tank having a circulating flow of warm water containing a weak alkali such as sodium carbonate and soap. The temperature of the water and the supply of alkali should be kept within specified limits, as the wool becomes too harsh if there is an excess of alkali or the temperature is too high. The wool in the bowls is kept moving by a series of prongs or forks which are

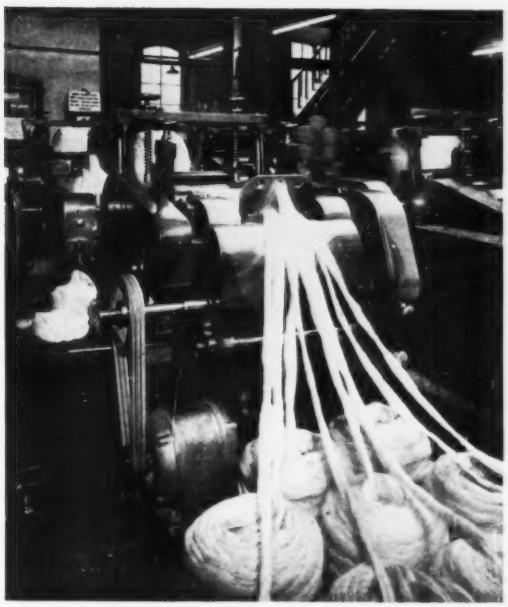


Fig. 3.—Blending gill boxes driven by 1½ h.p. 1,455 r.p.m. totally enclosed type ALK. 2 motors at Bond Street Mills

operated eccentrically. After each bowl there is a set of squeeze rollers which mangle the wet masses of wool emerging from the water. After the washing or scouring process, the wool contains about 1% of natural oil or grease and about 36% to 40% moisture. From the last squeeze rollers the wool is passed over a conveyor belt to a drying chamber which consists of three or four tiers of moving plates upon which the wool is carried. Steam pipes are fitted inside the drying chamber and fans circulate the hot air. When the wool leaves the dryer it contains about 20% moisture. Due to the low grease content, vegetable oil is sprayed on to the wool as it leaves the drying chamber, raising the grease content to about 3.5%. This makes the wool more pliable and at the same time promotes better carding properties.

The next process is carding and combing. The worsted carding machine comprises a series of revolving cylinders which are covered with fine wire spikes. The material is fed between the cylinders, and the action of the machine is to ' parallelise ' the fibres by the combing action between the moving drums, at the same time removing about 90% of the remaining vegetable matter. Before entering the machine the material is a tangled mass of washed wool, but when it leaves the machine it is in the form of a smooth sliver or rope-like mass. These slivers are fed into cans or coiled into balls.

From the carding machine, twenty or more slivers are combined and passed through a backwasher where they are thoroughly washed and rinsed, thus removing the remaining impurities. They are then passed through a drying chamber and thence to the ' backwasher gill box ' which combs the dry slivers and at the same time lubricates them to promote better spinning. They are again combed and passed through two further sets of gill boxes (strong boxes) and thence to a ' punch box ' where they are wound into balls, typically as shown in Figs. 3 and 4.

Up to this stage the slivers contain the long and short fibres, and for the production of worsteds these short fibres have to be removed. This process is carried out by a machine known as a Noble Comb which automatically separates the long fibres or ' tops ' from the short fibres or ' noils.' The tops are used for worsteds and finer quality hosiery wools, whilst the noils are used for blankets and similar material which is spun on the ' woollen ' system.

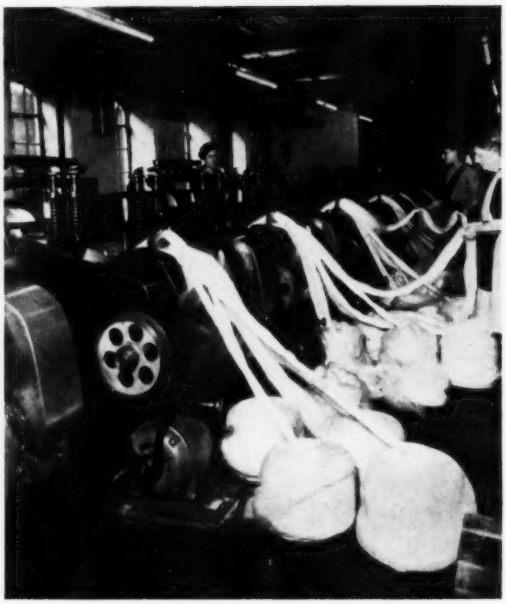


Fig. 4.—*Carding gill boxes individually driven by* 1 *h.p.* 955 *r.p.m. totally enclosed type ALK.* 3 *motors at Bond Street Mills*

The tops in the form of slivers are then passed through two sets of ' finisher gill boxes ' which completes the combing process. The slivers in the form of balls are then sent to the drawing department.

Drawing, as the name implies, is the drawing or attenuating of the slivers, reducing them in diameter to a small ' roving ' thread suitable for spinning effectively into a fine yarn. This reduction of sliver diameter is effected by passing combinations of slivers through a series of draw boxes. The slivers are drawn out at each successive stage and then wound on to bobbins. Many types of draw boxes are used for processing wool and to describe them in detail is beyond the scope of this article. Two systems of drawing are employed at Fielding & Johnson's Mills, however, namely ' Open drawing ' at the Bond Street Mills and ' French drawing ' at the Anker Mill. Some examples of the machinery used may be seen in Figs. 5, 6 and 7.

Fig. 5.—French drawing department at Anker Mill, showing 4 h.p. 1,440 r.p.m. totally enclosed fan cooled motors type LJ. 2/21 driving 44-spindle drawing frames

The fine roving which is produced in the drawing departments is then taken to the spinning frames. Three methods of spinning are employed by Fielding & Johnson, namely, mule, flyer and cap spinning. The principle of spinning is to draw out and twist the roving thread and produce a yarn of the desired thickness or ' count '. In the case of mule spinning (Fig. 8) the roving threads are drawn through a series of drafting rollers by means of a moving carriage which accommodates the spindles. As the carriage moves away from the rollers the yarn is drawn out and twisted, and is wound on to the spindles as the carriage returns to its starting position, after which the cycle of operations is repeated.

Fig. 6.—Individual drives to 30-spindle roving frames at Anker Mill, using 2½ h.p. 960 r.p.m. totally enclosed fan cooled motors, type LJ.2/21

The drawing and twisting actions for spinning yarn on a flyer frame are achieved by means of a fork mounted on a rotating spindle. The fork or flyer is provided with looped ends for guiding the thread on to the bobbins. As in the case of mule spinning, the threads are passed between a series of rollers and the combined action of the drawing out or drafting which takes place between the respective pairs of rollers rotating at different speeds results in the production of a fine thread.

In cap spinning (Fig. 9) the flyer is replaced by a cap surrounding the bobbin ; otherwise the process is similar to flyer spinning.

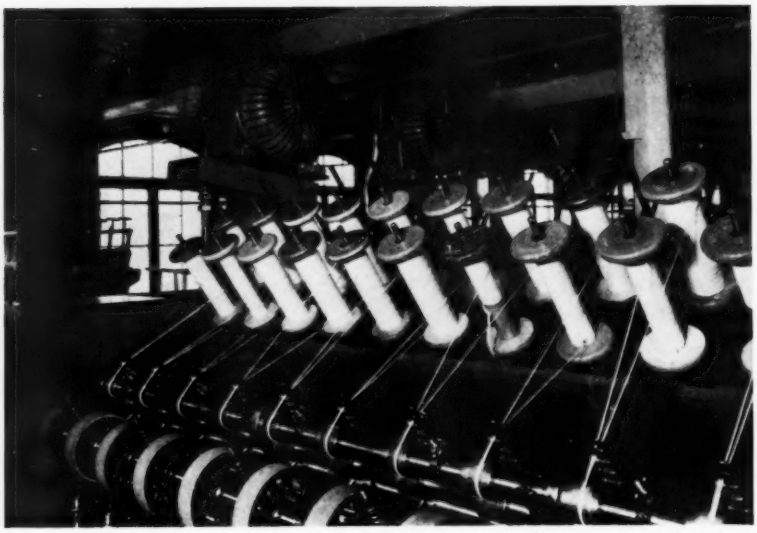


Fig. 7.
Roving frames at Bond Street Mills, driven by 2½ h.p. 1,455 r.p.m. totally enclosed motors

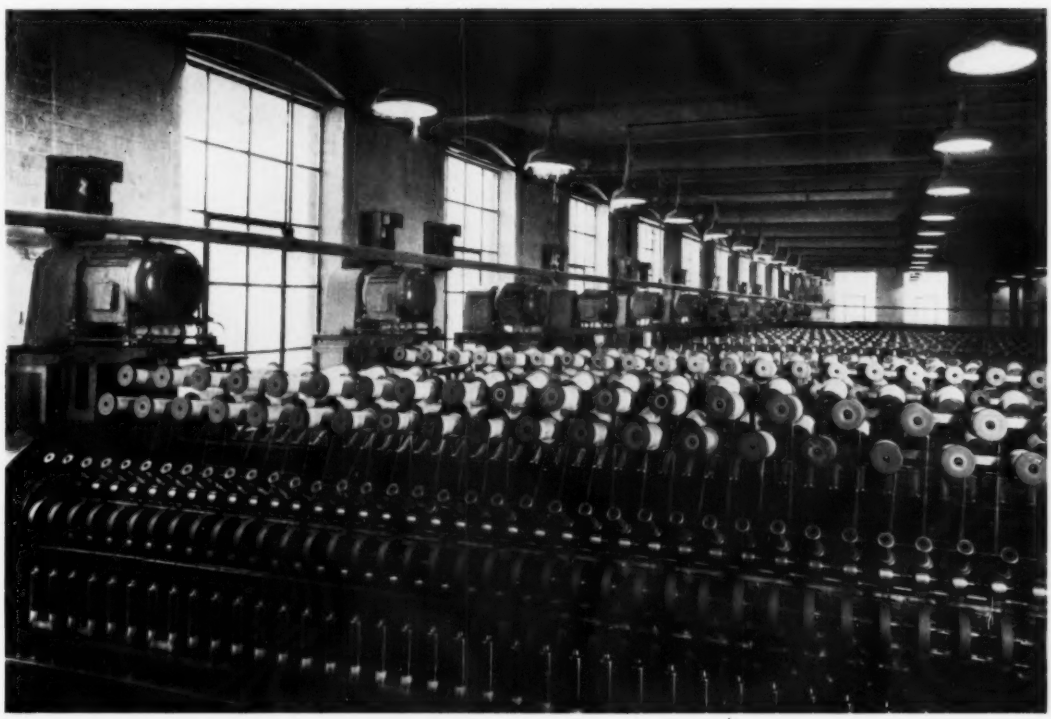
Fig. 8.—Mule room at Anker Mill, showing individual motor drives

Conversion to Electric Drives

In a straight conversion to individual motor drives one of the biggest problems is to modify machinery originally designed to be driven by flat belts from overhead lineshafting. Most machinery driven by this means is provided with the conventional fast and loose pulleys or a form of clutch pulley for taking the flat belt from the shafting above. The problem is to accommodate an electric motor either within the frame end or on brackets above the frame of the machine, and typical examples of conversions can be seen in Figs. 3 to 9 inclusive. Also it is preferable to dispense with the flat belt pulleys and use a suitable Vee belt transmission, taking into account restrictions in pulley diameters and centre distances between the driving and driven shafts. It is of the greatest importance that the motor unit should be made, if possible, an integral part of the textile machine itself. Floor space in most textile mills is at a premium, and it is essential that the motor units should not be erected in alleys or passages adjacent to the machines. The mechanical adaptations for individual drives often call for expensive millwrighting, and it is therefore important to modify the machinery as little as possible on this account.

One of the difficulties encountered in replacing a steam engine is to preserve a balance on the transmission shafting to ensure that the load on the prime mover is kept under complete control during the period of conversion. Consequently only the lightly loaded portions of the mill can be changed over in advance, leaving the major part to be dealt with during a holiday shut-down. To avoid any loss in output it is therefore essential that the individual drives should be installed ready, leaving only the Vee ropes to be fitted.

Fig. 9.—Individual drives to 220-spindle cap spinning frames, by 10 h.p. 1,455 r.p.m. totally enclosed fan cooled type LJ./21 motors, at Anker Mill



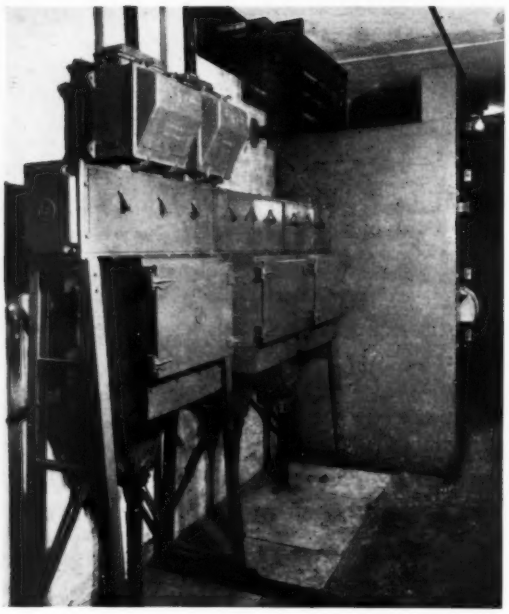


Fig. 10.—*Left-hand section of main low-voltage air-break switchboard at Bond Street Mills*

After the experience obtained in having the engines dismantled at Bond Street Mills, Fielding & Johnson decided to dismantle the engines at Anker Mill themselves. For this purpose an overhung saw was built by their engineering department, provided with a 2 ft. blade and driven by a standard 2 h.p. 'English Electric' motor via a Vee-rope double reduction drive. This saw was employed for cutting up the heavier parts of the engine, i.e. beams, connecting rods and second motion shafts etc., and care had to be exercised during dismantling due to the weight of the flywheel (40 tons) and the necessity for keeping the various shafts balanced.

Bearing in mind the processes in worsted mills already described, and the problems involved in converting an engine driven mill to individual motor drives, the importance of the electrical supply and distribution systems in converted mills will be apparent.

The electricity supply for the Bond Street and Abbey Mills is taken from the East Midlands Electricity Board, Leicester, at 415 volts 3-phase 50 cycles, and for the Anker Mill from the East Midlands Electricity Board, Nuneaton, at 6.6 kV 3-phase 50 cycles, which is reduced to 400 volts by two 500 kVA transformers. At each mill a main low-voltage switchboard is installed comprising incoming type 'OB3' air circuit-breakers with 'Combination' fuse-switches for the outgoing feeders. A section of the switchboard at the Bond Street Mills is shown in Fig. 10.

Careful study of the disposition of the productive machinery within the mills, and the sectionalising of motor circuits, resulted in distribution schemes of the most economical character, designed to reduce to a minimum the possibilities of stoppages in the various sections of production flow. The power and lighting loads are segregated, and each outgoing feeder is connected to a section board consisting of wall-mounted 'Combination' fuse-switch units on a common busbar chamber, which in turn feeds distribution boards for the various motor drives. The L.V. distribution panel in the air conditioning room at Anker Mill, incorporating 'Combination' fuse-switch units, is shown in Fig. 11. Provision has been made for metering the supply to each department for accountancy purposes.

Individual Electric Drives

The benefits of individual drives for textile machinery are now generally appreciated. Line-shaft drives and attendant flat belt transmissions, besides reducing the overall operating efficiency of a mill, are always a potential source of danger to the operative. Most textile machinery is admirably suited for individual electric motor drives, and its design has not altered basically during the past half century, although in recent years great progress has been made in the field of textile electrification and much knowledge has been gained of the operating characteristics. The power requirements of certain types of machinery vary considerably; each machine which is to be driven by an individual motor has to be carefully studied on its own merits, and an intimate knowledge of the process is essential before selecting the driving unit.

Most of the preparatory machinery including the scouring, carding, combing and drawing plant

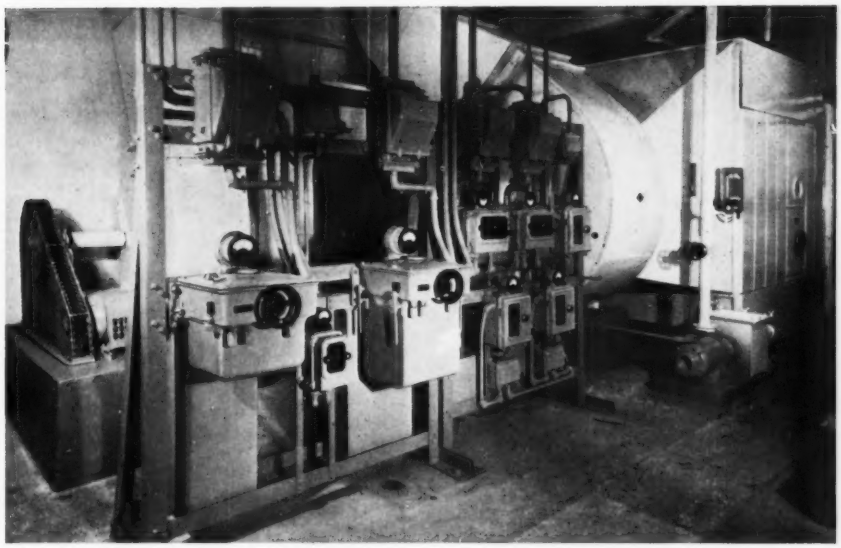


Fig. 11. 'Combination' fuse-switch type low-voltage distribution units in the air conditioning room at Anker Mill

is subject to variable loading conditions. In addition the machines have heavy inertia loads, and careful attention has to be given to the selection of motors of suitable horse-power outputs and with starting torques to withstand the severest operating conditions. Such factors as acceleration times and electrical knock-off motions have also to be considered. In the spinning processes this is even more important, as the over-powering of such machinery will result in broken threads every time the motor is started from rest.

At Fielding and Johnson's Mills tests were carried out on a number of machines to determine motor starting torques and horse-power ratings. For the worsted mules at the Anker Mill, which are motorised individually, recording wattmeters and ammeters were used on an experimental prototype drive so that a careful study could be made of the load fluctuations. By this means, after considering the worst operating conditions, the correct motors and starters could be selected. Motor enclosure is another important factor. Most of the machines installed are of the totally enclosed fan cooled or straight totally enclosed types, which are well suited to the dusty working conditions. The English Electric Company has supplied over 750 motors totalling about 3,500 h.p. for these mills.

Although, due to fluctuations in the cost of coal, electricity and maintenance during the past few years, it is very difficult to assess the actual net savings effected by using individual electric motor drives, as far as operating conditions are concerned it has been estimated that for group drives the average lineshafting, belting and transmission losses are of the order of 20-25%. Another advantage of the individual motor drive is the absence of speed variation of the driven machine, which obviously affects the overall efficiency of the related process. In most mills the lineshaft speed fluctuation due to variation of load may be as much as plus or minus 5%. Other factors in favour of the individual drive are the abolition of belts from the ceilings, which contributes to safer and cleaner working conditions, the elimination of shadows caused by overhead drives, and the absence of belt slip.

The electrification schemes for Messrs. Fielding & Johnson's mills were prepared by The English Electric Company in conjunction with Messrs. Gent, Hurley and Orringe Limited, electrical contractors.

Depth Indicators for Mine Hoists

By H. DOLAN, M.I.Mech.E., A.M.I.E.E., Chief Engineer, Mining Division, and
A. SHAW, B.Sc.(Eng.), Mining Division.

FOLLOWING THE INTRODUCTION of intensive mining employing either vertical or inclined shafts, the need for the hoist driver to have visual indication of the relative position of the conveyance at all times became imperative. Expansion of the industry both in Great Britain and overseas led to the insertion of clauses in the relevant official regulations governing the use of depth indicators on mine hoists, and in the first issue of the Coal Mines Act of Great Britain in 1911, Section 40, Sub-section 10, stated:—

> "*. . . . and a proper indicator (in addition to any mark on the rope) showing to the person who works the machine, on a dial or in some sufficient manner, the position of the cage in the shaft and placed in such a position as to be easily seen by him at the same time as the marks on the rope.*"

In the same year a similar regulation was laid down by the Government Inspectorate of Mining Machinery in South Africa. This regulation as issued in the Mines, Works and Machinery Regulations of South Africa reads in its present form:—

> "*Every winding engine shall, in addition to any marks on the rope, be provided with reliable depth indicators conveniently situated, which will clearly and accurately show to the engine driver at his driving seat at all times:—*
>
> (*a*) *The position of the cage, skip or other means of conveyance; and*
>
> (*b*) *At what places in the shaft changes of gradient necessitate reduction in speed.*
>
> *On any new engine erected after 1st October, 1930, the pointer of the dial indicator on the driver's right hand shall move in a clockwise direction when lowering, and in the case of a post and spiral indicator the pointer shall move up or down as the conveyance moves up or down.*"

Several types of depth indicators have been used in an endeavour to obtain an accurate indication of the position of the conveyances in the shaft, and in the following article brief consideration is given to the various designs. Examples of all these types, many of which are still manufactured, can be found on existing hoist equipments.

Vertical Screw Type

Imitation of the vertical movement of the conveyances in the shaft led naturally to the idea of a vertical screw depth indicator comprising single or twin vertical screws carried in bush bearings and driven from the hoist drum or respective drums by means of chain and/or bevel gears and shafting. Each screw is fitted with a travelling nut carrying a pointer moving over a vertical scale on which marks can be made to represent the various levels. The weight of the nut always bearing downwards on the screw tends to reduce the effect of backlash in the indicator itself. An approach warning bell and overtravel limit switches are generally fitted, operated by the travelling nut. If the hoist is fitted with clutched drums for multi-level winding or rope adjustment, two independently driven vertical screws are necessary, each being driven from its respective drum. Generally the height of such indicators varies between four and eight feet according to the depth of shaft and accuracy of indication required, and typical examples are shown in Figs. 1 and 2.

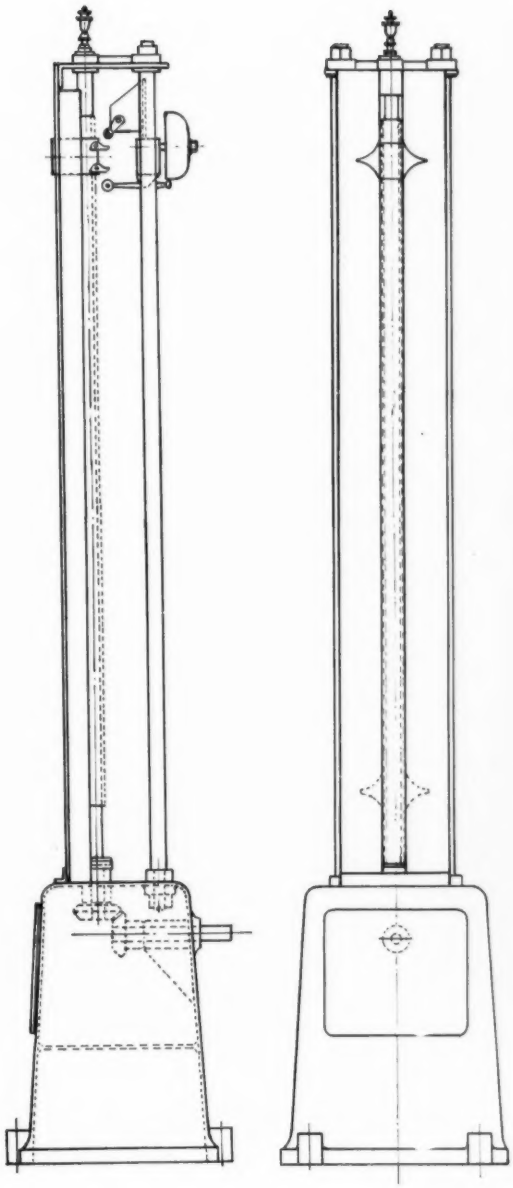


Fig. 1.—General arrangement of vertical screw type depth indicator

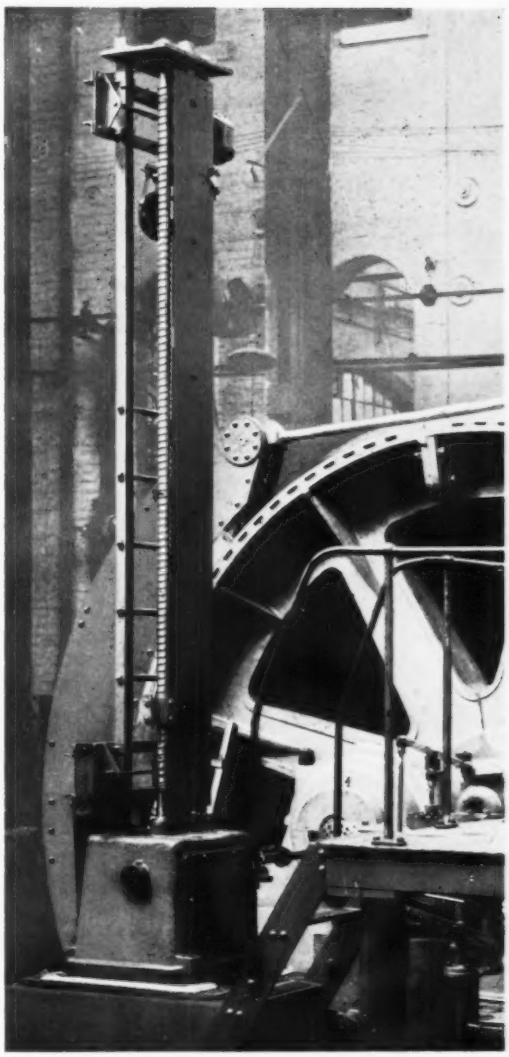


Fig. 2.—A vertical screw type depth indicator during erection

Circular Type

With the advent of deep level mining it became apparent that to obtain a reasonable degree of accuracy the vertical type would become physically unwieldy and hence the circular type is considered more favourable. It is obvious that the circular dial has the advantage over the vertical screw,

since approximately three times the scale length can be obtained for the same diameter of dial as vertical screw length.

In its simplest form this type, illustrated in Figs. 3 and 4, comprises a gear driven pointer moving over a circular metal plate on which marks can be added representing the various levels. To reduce backlash a counter weight is fitted to keep mating teeth of the bevel gears in contact within the indicator. An approach warning bell and overtravel limit switches can be fitted as with the vertical screw indicator. Similarly it is necessary with clutched drum equipments to have independently driven pointers.

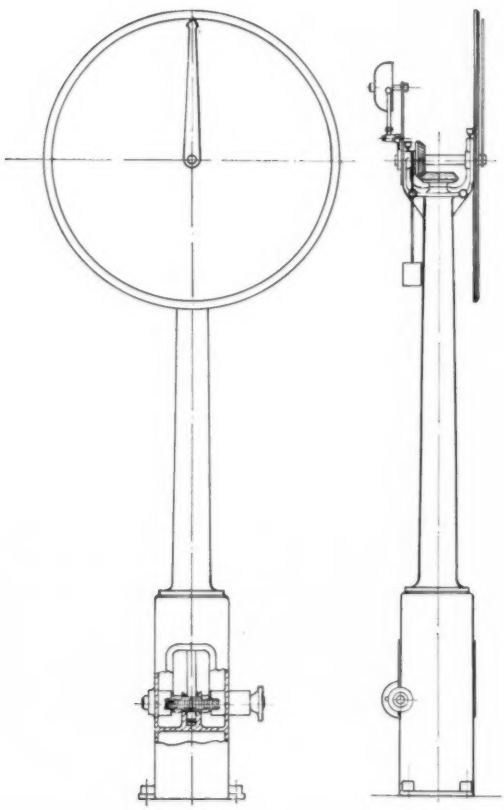


Fig. 3.—General arrangement of circular type depth indicator

Both the aforementioned types, whilst being simple and comparatively cheap to manufacture, are not of any assistance to the driver in landing the conveyance at the correct level, and it is therefore essential to have chalk marks on the drum in order to obtain accurate decking. The physical positioning of these indicators in the engine house has generally been such that the driver's lines of vision to the indicator and the drum marks are not coincident.

To overcome the inability to give accurate landing level indication, further designs of depth indicators have been produced.

Vertical Screw Type with Additional Dial

One or two circular dials are mounted on top of the indicator having their respective pointers actuated by the upward movement of the appropriate travelling nut ; an example of this type is shown in Fig. 5. During the last turn of the hoist drum when the conveyance approaches the landing level, the dial pointers rotate approximately one complete revolution, thus giving a vernier effect. Whilst greater accuracy is obtained over the last drum revolution in this manner, limitations are imposed by the size of dial it is possible to accommodate. It will be realised that this additional pointer will only give accurate indication at the bank or dumping levels and not at any intermediate landing level.

Double Pointer Type

A circular dial type having two indicator pointers was designed by McCulloch, as indicated in Fig. 6, both pointers being mechanically driven from the main drum shaft and arranged by gears so that one pointer moves 1/20th while the other moves 1+1/20th of a complete revolution for every 100 ft. traversed in the shaft. The pointers are therefore always coincident at a position corresponding to 100 ft. of vertical depth. The advantage of this design over the ordinary circular dial type cannot be fully realised unless all the intermediate levels are suitable distances apart, *i.e.*, exact multiples of 100 ft. Also, since rope stretch is bound to occur during the life of the rope, it is obvious that disparity of level indication must occur since the pointers of the indicator are driven

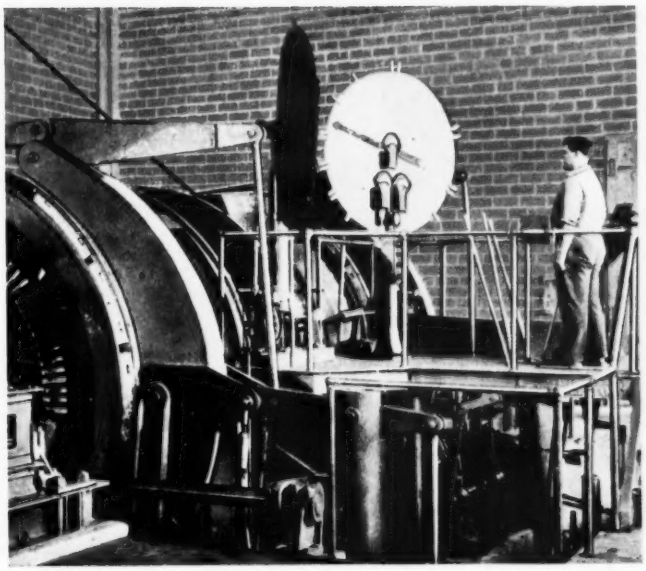


Fig. 4.—Example of a circular type depth indicator

from the hoist drum, thus defeating the advantage claimed for this design.

Barrel Type

This type is illustrated in Figs. 7 and 8 and consists of a cylindrical drum with separate vertical screws, the drum and the screws being mechanically driven from the main hoist drum. The indicator drum can be arranged to make one complete revolution per hoist drum revolution or for say 100 ft. vertical distance in the main shaft. Pointers carried on the vertical screws move up and down a distance equal to a spiral groove pitch for each barrel revolution, indicating relative movement of the conveyances. As the barrel rotates whilst the pointers move vertically, the effect is to trace a spiral on the indicator drum. By this means a longer effective scale length can be obtained than with the vertical screw or dial indicator of comparable physical dimensions. The relevant level marks are placed on the spiral and the correct position for decking is given by the coincidence of this mark and the extremity of the vertical moving pointer. Owing to the construction of such an indicator the marks on the spiral are only visible for a quarter of a revolution before the landing level is reached, and this constitutes a serious disadvantage of this design.

Improvements and Developments

Attempts have been made to improve the large vertical depth indicators with a view to eliminating the screws and overcoming inaccurate indication due to wear of moving parts having a large inherent inertia.

One particular improved vertical model, Fig. 9, is that in which the vertical screws and travelling nuts are completely replaced by an endless chain on which are mounted

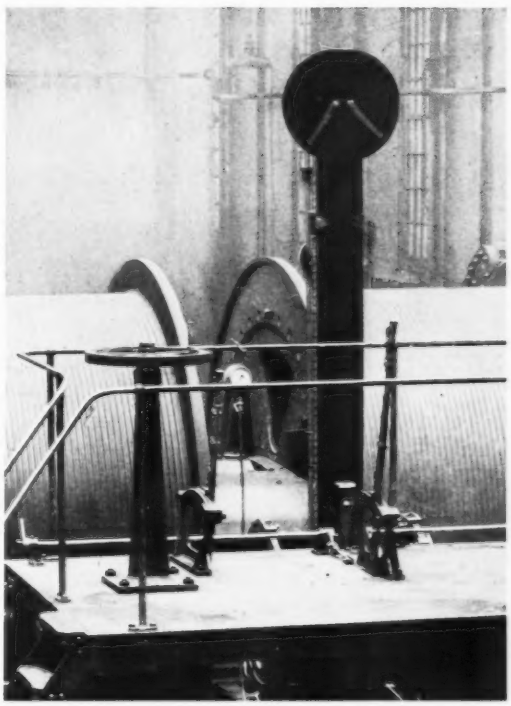


Fig. 5.—Vertical screw type depth indicator with additional dial

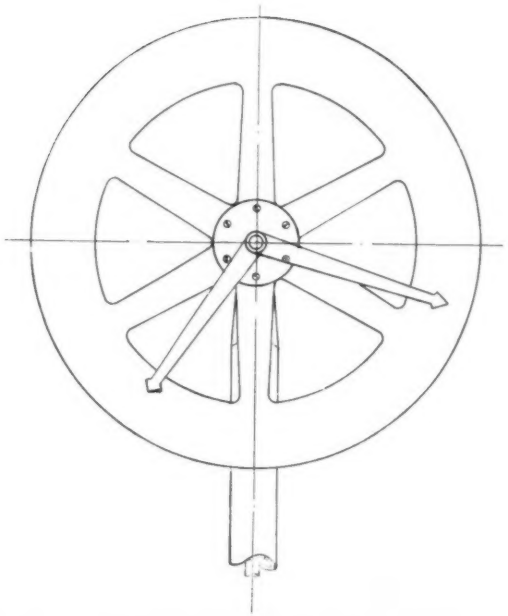


Fig. 6.—Double pointer type depth indicator.

(From "Winding Engines and Winding Appliances," by McCulloch and Futers, with kind permission of the publishers, Messrs. Edward Arnold & Co.)

light-weight pointers. These pointers travel up and down over an illuminated panel to give indication of the position of the conveyances in the shaft. In addition a circular dial and pointer are provided, so arranged that the pointer only rotates during the last drum revolution before decking or dumping of the conveyances.

The relative physical position in the engine house of any type of large indicator is critical in order to prevent obstruction of the driver's view and yet retain the necessary angle of vision.

With the introduction of desk control it was considered that the depth indicator or indicators should be mounted with the other instruments on the desk. It was desired that such miniature depth indicators for mounting on control desks should fulfil three main conditions, namely, that they should be:—

(i) At least as accurate as the large types to which reference has previously been made, especially during the approach to the landing level.
(ii) Compact for mounting on the control desk.
(iii) Sufficiently illuminated without causing glare.

One of the first attempts to meet these conditions was the development of a clock type indicator having a slow moving pointer passing over

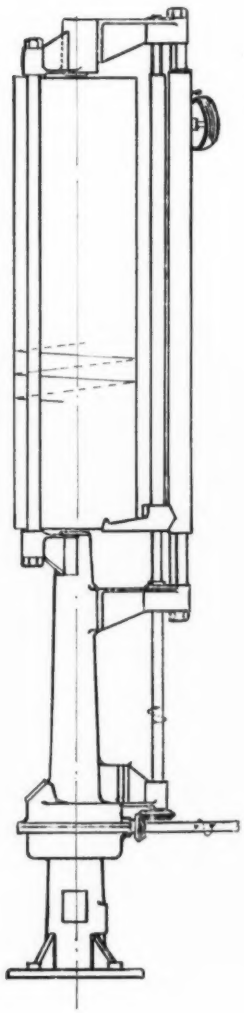


Fig. 7.—General arrangement of barrel type depth indicator

a scale calibrated in drum revolutions, and two further smaller pointers and scales inset into the large scale. These small pointers, one for each conveyance, rotate at drum speed, their associated stationary scales being calibrated in feet. These small scales are intended to be used as verniers to obtain accuracy at decking or dumping level. The disadvantages of such a type are the reduced diameter of the vernier scales and the associated pointers continuously revolving at hoist drum speed in front of the driver.

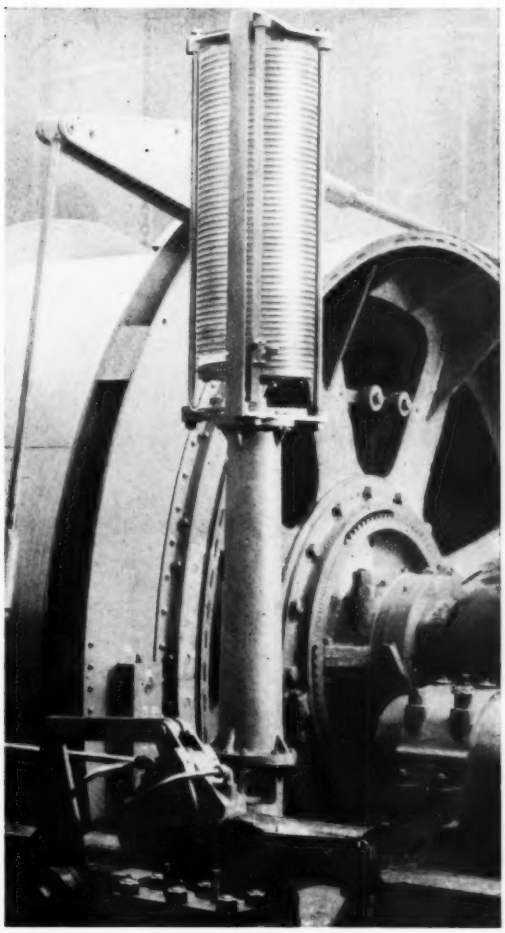


Fig. 8.—A barrel type depth indicator

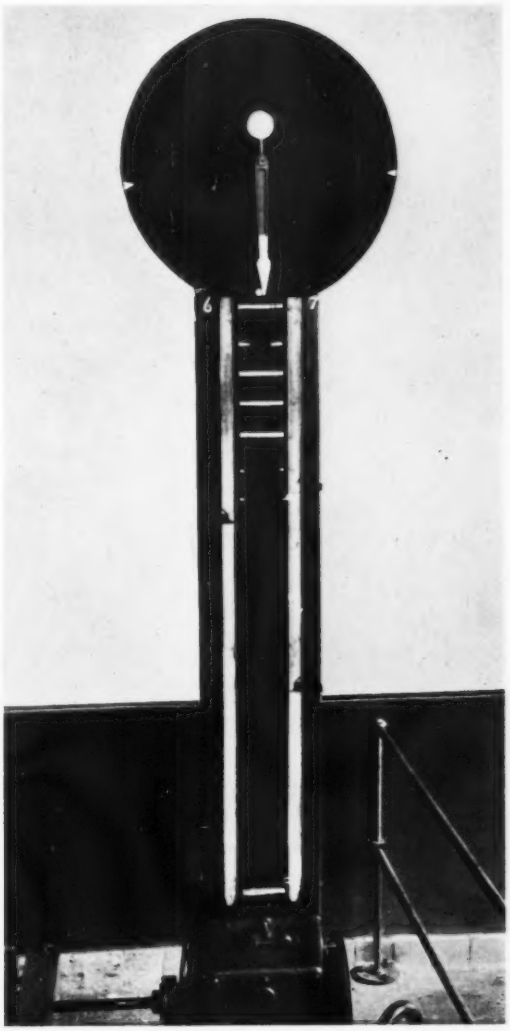


Fig. 9.—Improved vertical type depth indicator with additional circular dial

Another design was of the miniature barrel type, but this had a more marked disadvantage than the conventional large type due to the great reduction in barrel diameter.

For decking purposes chalk marks are placed on the drum cheek, and their position changed as necessary to allow for rope stretch. Such marks are

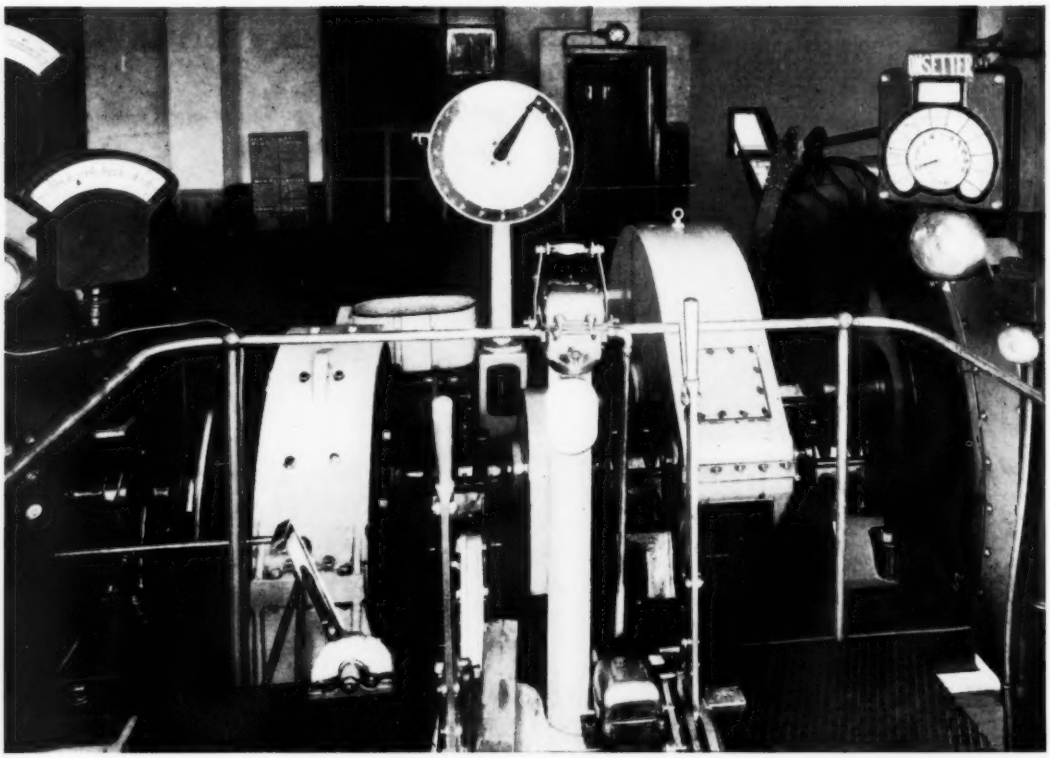
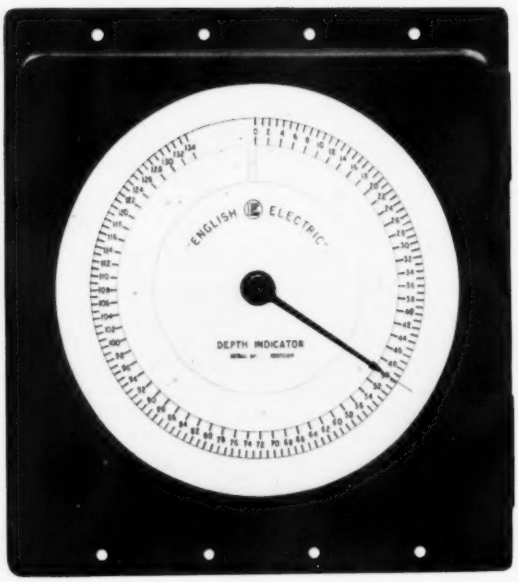


Fig. 10.—(*above*) *View from driver's position*

Fig. 11.—(*right*) *Dial of miniature depth indicator*



naturally only of use during the last drum revolution before the final landing is reached, and are therefore used as a vernier scale to the conventional large depth indicators. Due to the physical positioning of these large indicators, in practically all cases the driver has to change his line of view from the indicator to the drum cheek. In the majority of cases the line of view is practically tangential to the plane of the drum cheek, making it difficult to observe the chalk marks for more than about a quarter of a revolution in either direction of rotation. The full implication of such an undesirable layout will be realised from Fig. 10, in which the disposition of the depth indicator, the instruments and signal indicators relative to the driver and drum cheek involve four different lines of view.

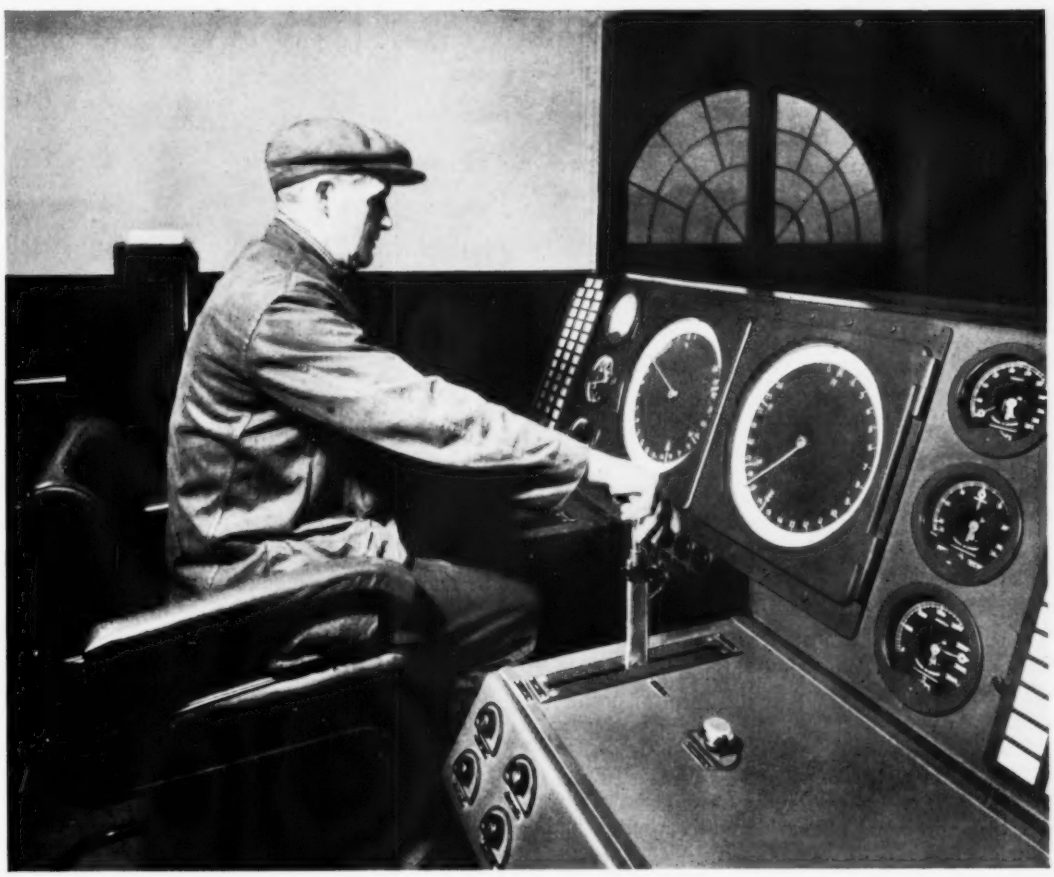
The English Electric Company first patented and developed a miniature depth indicator in 1942. The principle underlying this basic design of depth indicator for mounting on control desks was to combine in a single unit the circular dial of the position indicator with an imitation drum cheek in the same plane, as depicted in Fig. 11. The circular vernier scale (imitation drum cheek), gear driven to rotate at the hoist drum speed, can be marked in a way corresponding to that adopted for the drum cheek, but the decking marks on the vernier scale are completely visible throughout the whole of the last drum revolution.

Another advantage of this design is the elimination of the fast moving vernier pointer. With such an indicator (Fig. 12) mounted within 2 ft. 6 in. of the driver's eyes, it has been proved possible to obtain the accurate decking demanded when hoisting with tubs or mine cars in cages.

'English Electric' Miniature Type

A twin unit of the type now manufactured is illustrated in Fig. 13. Such units are supplied on double drum equipments where one or both drums are clutched. Where the hoist has only one rope drum, a single indicator unit only is required, and considerable numbers of this type are being made for drum and Koepe pulley hoists.

Fig. 12.—*Miniature depth indicator on control desk*



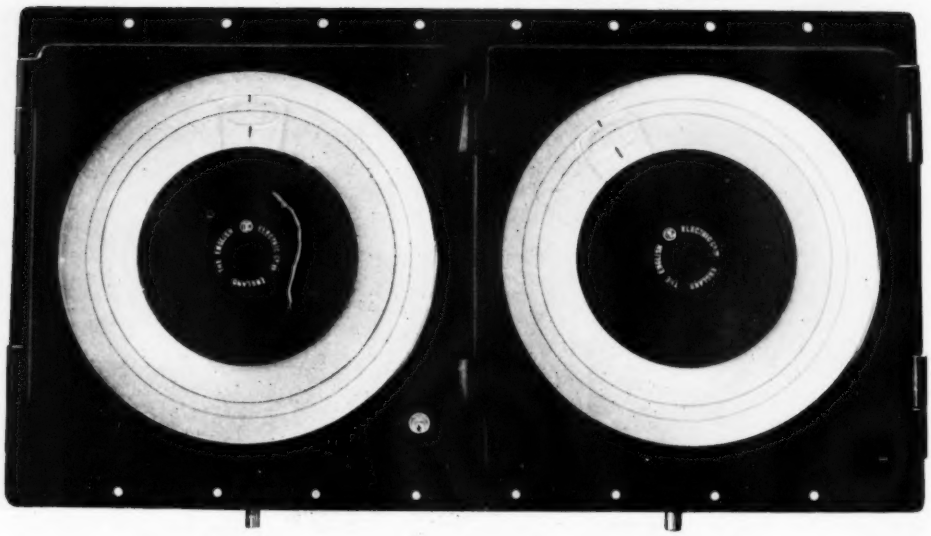


Fig. 13.—*Latest design of miniature depth indicator dials*

The following description gives details of this indicator which is the only miniature type yet permitted by the Government (South Africa) Inspectorate of Mining Machinery.

This depth indicator, patented and manufactured by The English Electric Company, is a precision instrument. It incorporates in the same plane an inner fixed dial, a vernier dial rotating at hoist drum speed concentric with the inner fixed dial, and an outer fixed dial surrounding the rotating vernier dial. An indication of the position of the conveyances in the shaft is obtained by means of a line-marked transparent pointer, the total angular rotation of which is approximately 315° for any shaft depth. Whilst this form of position indication is satisfactory between levels, the hoist driver requires to know the position of the conveyance much more accurately as the landing level is approached. To achieve this accuracy and enable the driver to land correctly, the vernier dial is incorporated which makes one complete revolution for each complete drum revolution.

Opal yellow perspex is used for the dials, which are not graduated. The pointer is made of clear perspex and a black line is engraved on the underside. Since the dials are left clear, the hoist driver may mark them with a special pencil to give the relation between the dials and the pointer corresponding to a particular position of the conveyance in the shaft. Any marks so made can easily be removed when required with a dry cloth.

For twin units the pointer on the driver's right rotates clockwise when *lowering* the right hand conveyance. The pointer on the driver's left rotates clockwise when hoisting the left hand conveyance. Both pointers thus rotate in the same direction at the same time. The pointer of the indicator for single units is geared to rotate clockwise when lowering the right hand conveyance.

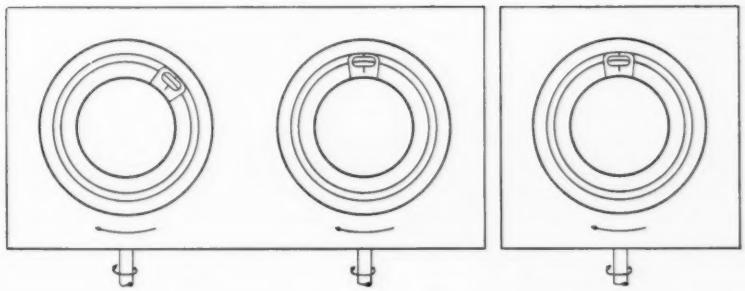


Fig. 14.—Standard pointer rotations for single and twin unit indicators when right hand conveyance is lowering

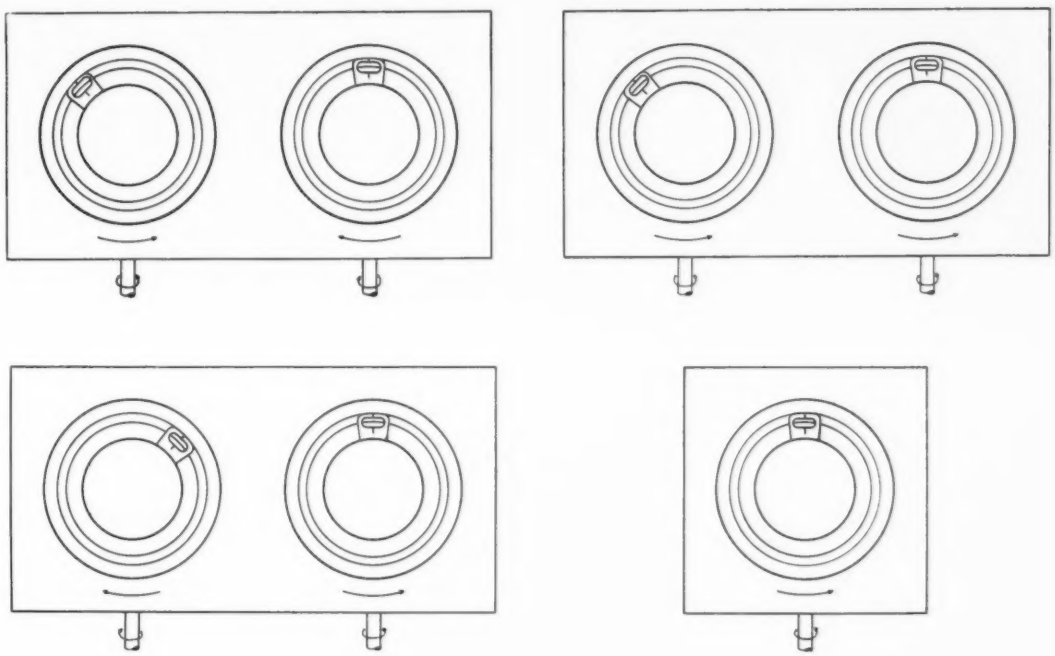


Fig. 15.—Alternative pointer rotations for single and twin unit indicators when right hand conveyance is being lowered

These arrangements conform to the regulations in South Africa and to the generally accepted practice in Great Britain and other countries.

In a twin unit the two sets of gearing within each indicator are identical, and for all twin or single units the main drive shaft to the indicator rotates anti-clockwise when viewed from the underside for clockwise rotation of the pointer, the external mechanical drives from the hoist drums being suitably arranged, see Fig. 14. Alternative pointer rotations can be given, as illustrated in Fig. 15, if preferred. Each indicator is provided with a substantial shaft extension supported in ball journal bearings.

When the indicator unit is mounted on the sloping face of a control desk, a special universal coupling is provided for mechanical connection to the individual drive from the hoist drum, the

indicator shaft being keyed to receive this coupling.

To facilitate accurate setting and adjustment of the indicator (as is most specially required for Koepe pulley hoists), a vernier coupling is mounted either between the universal coupling and the main drive in the desk (see Fig. 16) or externally. The main drive rotates at the hoist drum speed, the vernier dial supported in ball journal bearings being driven through bevel gearing at the same speed. Backlash is eliminated in the gearing by splitting one of the bevel wheels into two sections of teeth which are spring loaded so as to thrust on opposite sides of the mating teeth of the solid driving bevel. The rotating pointer is worm wheel driven from a further set of bevel wheels (in which backlash is again eliminated) from the main drive shaft, and the plain sleeve bearings in which the pointer shaft runs have Oilite bushes (see Fig. 17). Gears associated with the indicator are of phosphor bronze, and before leaving the Works all bearings are packed with special grease which, as the indicator is dust proof, will provide satisfactory lubrication for at least two years in an ambient temperature not exceeding 40°C.

For transportation the whole indicator movement is locked with the pointer set within its designed angular travel to prevent damage to the backlash arrangements. It is also essential that the pointer should not be allowed to rotate beyond its designed limit of travel, and temporary marks are added indicating the maximum travel limits.

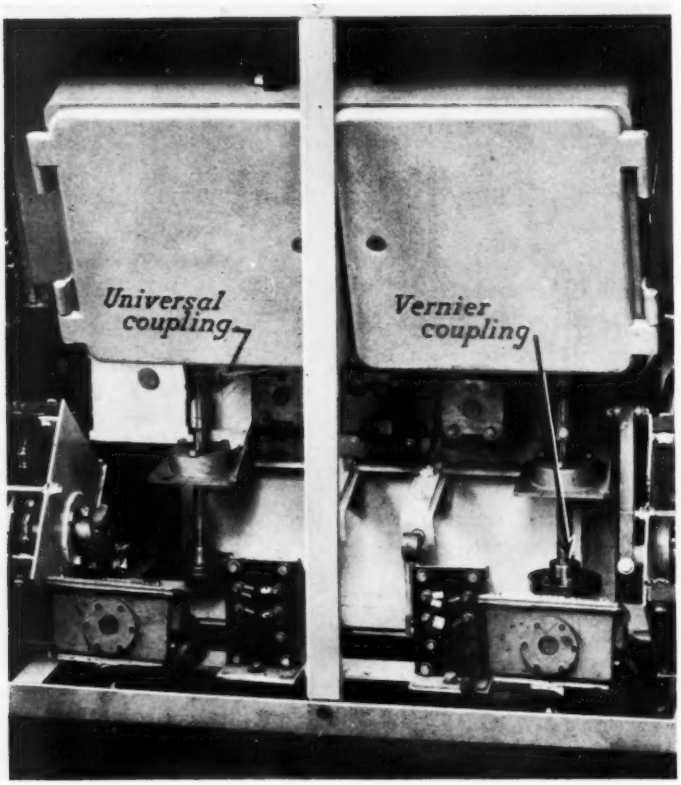


Fig. 16.—Vernier and universal couplings in 'English Electric' control desk with sloping face

Indirect illumination of the indicator dials of the unit is employed, and an even intensity is obtained by means of a specially shaped matt white reflector (Fig. 18) situated behind the dials. Three lamps are used connected in parallel, which are symmetrically disposed about the centre spindle, all lamps being accessibly arranged for easy replacement from the front.

The case is of cast alloy to combine lightness with mechanical strength, and is provided with flanges to facilitate mounting on a control desk or equivalent equipment directly in front of the hoist driver.

Anti-slam devices are fitted to the glass-windowed front door of each indicator and a sealing gasket of circular section is partially inset in the alloy door frame. The door is fastened by two handles fitted with shaped internal locking plates, ensuring compression of the gasket against the machined face of the housing. This prevents the ingress of dust and moisture. The two door

Fig. 17.—*Internal gear drives of miniature depth indicator*

handles are interlocked and provided with a single flush type Yale lock. In the case of twin units the handles are set between the two doors which open outwards from the centre line of the assembly.

The rear door of the indicator is also hinged and similarly fitted with a non-perishable sealing gasket. In view of the greater inherent rigidity of this door, a single flush type Yale lock is fitted. Each rear door of a twin indicator unit is individually locked (see Fig. 19).

To obviate halation the whole unit is finished in a matt colour, usually eau-de-nil.

This miniature type of precision indicator has the following advantages not previously afforded by other types of indicators :—

(i) Complete vision of the vernier dial (imitation drum cheek) for the whole of the last hoist-drum revolution when approaching the landing level.

(ii) The exceedingly close proximity of the indicator to the driver makes it possible to read with great accuracy.

(iii) The driver is not required to change his line of vision during landing operations, thus relieving eye strain and general fatigue.

(iv) Equally accurate landing can be obtained at all levels in the shaft.

(v) Extremely low inertia of all moving parts.

(vi) Special backlash arrangements for both vernier scale and position pointer.

(vii) Concealed anti-glare illumination.

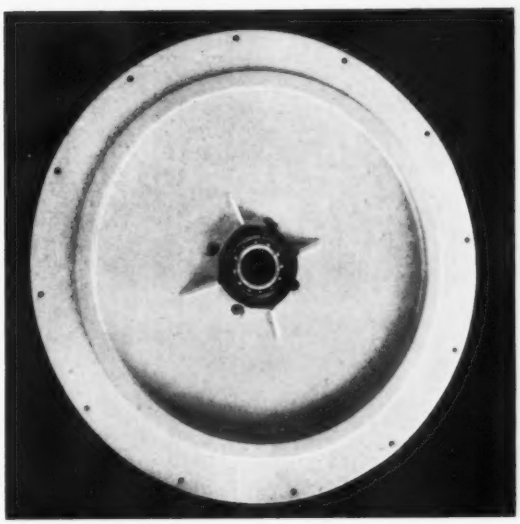


Fig. 18.—*Reflector of miniature depth indicator*

Comparative Accuracy

To illustrate the differences in the accuracy obtainable with the most common types of depth indicators the following calculations are included, the conclusions being summarised in Table I for comparative purposes.

The dimensions of the various indicators are typical for the shaft depth of 5,000 ft. taken, and essential data of the hoist used for the calculations are:—

Diameter of drums, 14 ft. 0 in.
Width of each drum (2 drums), 6 ft. 6in.
Rope diameter, $1\frac{3}{4}$ in.

From this data it is found that three-layer coiling is necessary, the average rope movement per drum revolution being 44.8 ft.

The large type depth indicators are assumed to be located a typical distance of 12 ft. from the driver, and the miniature precision type depth indicator is located 2 ft. 6 in. from the driver, being mounted on a control desk. The importance of this positioning is illustrated in Fig. 20, where a movement 'AB' on the large type depth indicator 12 ft. from the driver is equivalent to a movement 'ab' on the miniature type depth indicator 2 ft. 6 in. from the driver.

$$ab = \frac{2.5}{12} \times AB = 0.208\ AB$$

It has been found in practice that the driver can easily distinguish a $\frac{1}{4}$ inch separation of decking marks at 2 ft. 6 in. (equivalent to 1.2 in. at 12 ft.).

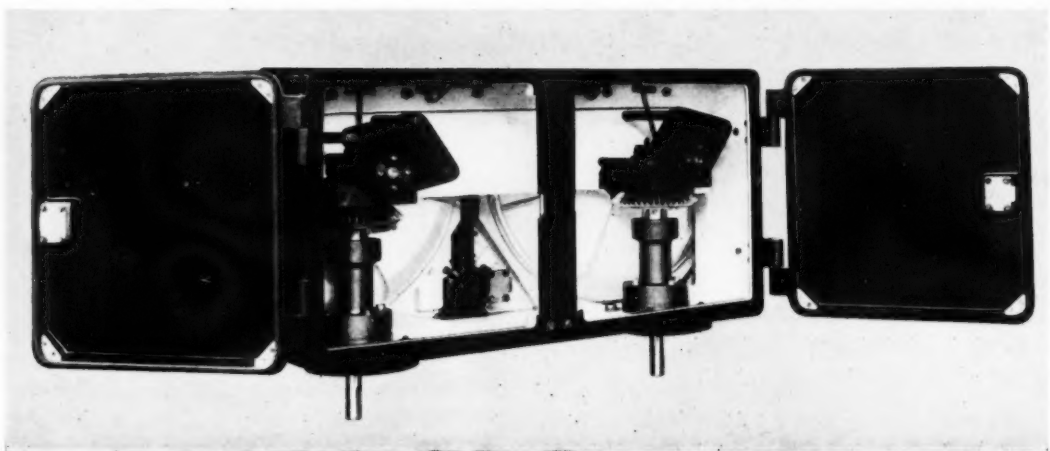
VERTICAL SCREW TYPE

Screw length .. .. =7′ 8″
Movement of pointer per foot of rope travel $= \frac{92}{5{,}000}$ .. .. =0.0184″ (AB)
Equivalent movement at 2′ 6″ from driver per foot of rope travel = 0.0184 × 0.208 =0.00383″ (ab)
Rope travel per $\frac{1}{4}$″ equivalent movement of pointer at 2′ 6″ from driver
$= \frac{0.25}{0.00383} = 65.3$ .. .. .. =65′ 3½″

CIRCULAR TYPE

Diameter of dial .. .. =8′ 0″
Total angular travel of pointer for 5,000 feet shaft depth .. .. =315°
Total pointer scale length $= \pi \times 8 \times \frac{315}{360}$ =22′ 0″
Movement of pointer per foot of rope travel $= \frac{22 \times 12}{5{,}000}$ .. .. =0.0529″
Equivalent movement at 2′ 6″ from driver per foot of rope travel=0.0529×0.208 .. =0.0110″

Fig. 19.—Rear view of miniature depth indicator



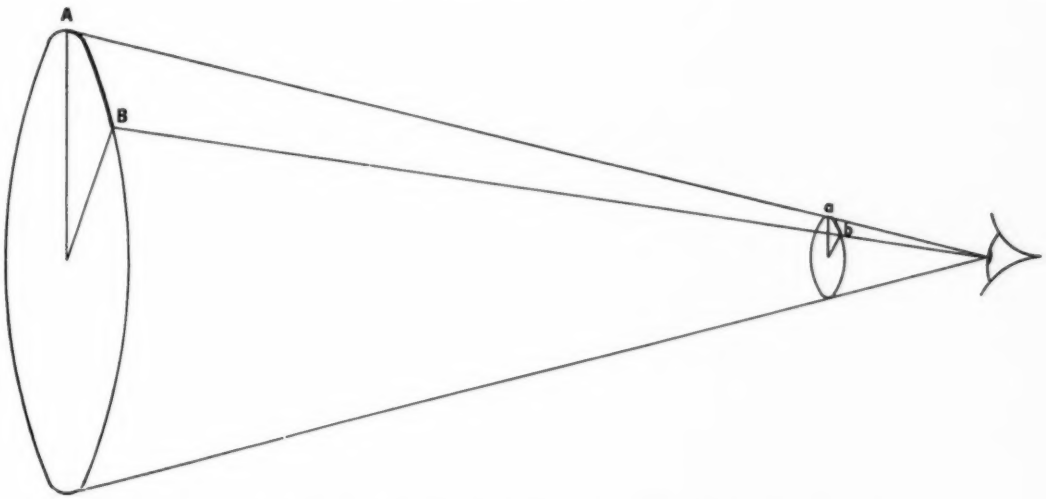


Fig. 20.—Relation between pointer displacement and position of indicator

Rope travel per $\frac{1}{4}''$ equivalent movement of pointer at 2′ 6″ from driver $= \frac{0.25}{0.0110} = 22.7$ .. .. $= 22'\ 8\frac{1}{2}''$

VERTICAL SCREW TYPE WITH ADDITIONAL DIAL

Screw length .. .. $= 7'\ 8''$
(In the middle of the wind, movements as for the vertical screw type.)
Diameter of dial .. .. $= 18\frac{1}{2}''$
Total angular travel of pointer during last drum revolution .. .. .. $= 300°$
Total pointer scale length $= \pi \times 18.5 \times \frac{300}{360}$ .. .. $= 48.5''$
Rope travel during last drum revolution .. $= 44.8'$
Movement of pointer per foot of rope travel $= \frac{48.5}{44.8}$ .. .. $= 1.08''$
Equivalent movement at 2′ 6″ from driver per foot of rope travel $= 1.08 \times 0.208$ .. $= 0.225''$
Rope travel per $\frac{1}{4}''$ equivalent movement of pointer at 2′ 6″ from driver $= \frac{0.25}{0.225}$
$= 1.11$ .. .. $= 1'\ 1\frac{1}{4}''$

BARREL TYPE

Length of barrel .. .. $= 4'\ 1''$
Diameter of barrel .. .. .. $= 18''$
Pitch of spiral groove .. .. .. $= 1''$
Number of barrel revolutions for 5,000 feet shaft depth .. .. $= 49$
Rope travel per barrel revolution $= \frac{5{,}000}{49} = 102'$
Movement of mark on spiral groove (barrel circumference) per foot of rope travel $= \frac{\pi \times 18}{102}$ .. .. $= 0.555''$
Equivalent movement at 2′ 6″ from driver per foot of rope travel $= 0.555 \times 0.208 = 0.116''$
Rope travel per $\frac{1}{4}''$ equivalent movement of pointer at 2′ 6″ from driver $= \frac{0.25}{0.116}$
$= 2.16$ .. .. $= 2'\ 2''$

MINIATURE PRECISION TYPE

Inside diameter of outer fixed dial .. $= 13\frac{3}{8}''$
Total angular travel of pointer for 5,000 feet shaft depth .. .. $= 315°$
Total pointer scale length $= \pi \times 13.375 \times \frac{315}{360}$ .. .. $= 36.7''$
Movement of pointer per foot of rope travel $= \frac{36.7}{5{,}000}$ .. .. $= 0.00734''$
Rope travel per vernier dial revolution .. $= 44.8'$
Movement of vernier dial per foot of rope travel $= \frac{\pi \times 13.375}{44.8}$ .. .. .. $= 0.937''$
Rope travel per $\frac{1}{4}''$ movement of vernier dial $= \frac{0.25}{0.937} = 0.266$ .. .. .. $= 3\frac{1}{4}''$

TABLE I

Summary of comparative accuracy obtainable with different types of depth indicators, based on a shaft depth of 5,000 feet, drum width 6 feet 6 inches, rope diameter 1¾ inches and drum diameter 14 feet.

Type of Indicator	Vertical Screw	Circular	Vertical Screw with additional dial	Barrel	Miniature Precision
Dimensional features	Screw length 92″	Dial dia. 96″, total angular travel of pointer 315°	Screw length 92″, dial dia. 18½″, total angular travel of vernier pointer 300°	Length 49″, dia. 18″, pitch of spiral groove 1″	Vernier dial dia. 13⅜″, total angular travel of pointer 315°
Distance of indicator from driver	12′	12′	12′	12′	2′ 6″
Movement of main pointer* per foot of rope travel	0.0184″	0.0529″	0.0184″	0.555″	0.00734″
Equivalent movement of pointer at 2′ 6″ from driver	0.00383″	0.0110″	0.00383″	0.116″	0.00734″ (actual)
Movement of vernier pointer or mark per foot of rope travel ..	None	None	1.08″†	None	0.937″
Equivalent movement of vernier pointer or mark at 2′ 6″ from driver	None	None	0.225″†	None	0.937″ (actual)
Rope travel per ¼″ movement (or equivalent) of decking or tipping pointer or mark at 2′ 6″ from driver	65′ 3½″	22′ 8½″	1′ 1¼″	2′ 2″	3¼″
Relative accuracy for decking or tipping	Assumed unity	3	59	30	240

* mark on spiral groove in case of barrel type. † during last drum revolution only.